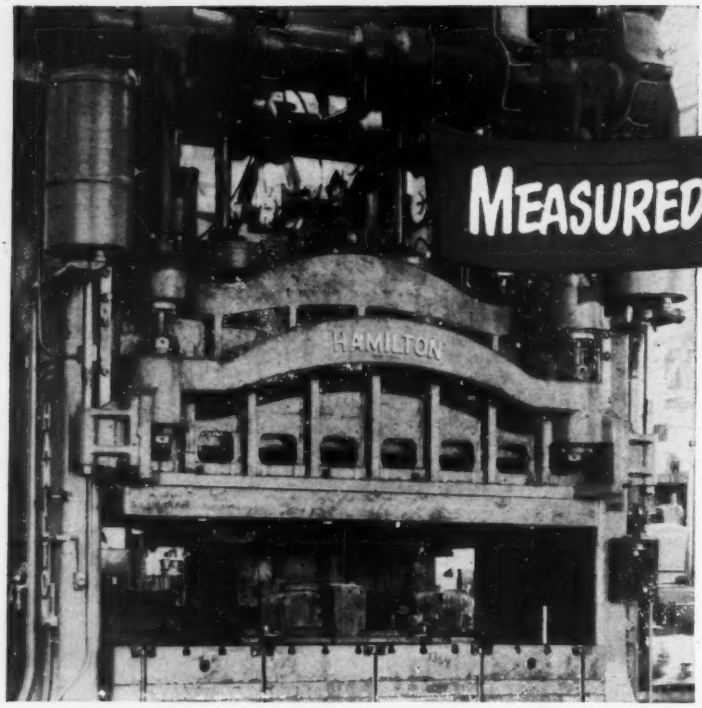
Lubrication ENGINEERING

JOURNAL OF
THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS



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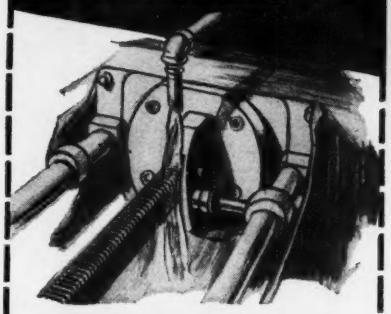

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DECEMBER, 1952, Vol. 8, No. 6

TABLE OF CONTENTS

Advertisers' Index 328
Annual Meeting Hi-Lites 280
Book Reviews 304
Coming Events 317
Committees, ASLE 325, 327
Current Literature 300
Directory, ASLE 320
Industrial Members, ASLE 322
Lube Lines 281
Lubricating Properties of Molybdenum Disulfide 285
by I-Ming Feng
Lubrication in the News 299
Lubrication Summarized 298
Material Specifications For Oil-Film Bearings 291
by E. B. Etchells
Membership Classifications, ASLE 324
Metal Cutting Developments 289
by R. J. S. Pigott
New Members 303
Patent Abstracts 302
Personals 304
Planned Lubrication As A Part of Plant Maintenance (Practical Lubrication, No. 4) 282
by T. R. Witt
Role of the Lubricant in Bearing Fatigue 295
by E. S. Carmichael and R. B. Purdy
Section News 301
Subject and Author Indexes, 1952 305
Tired Bearings—Reasons and Remedies 293
by J. Palsulich

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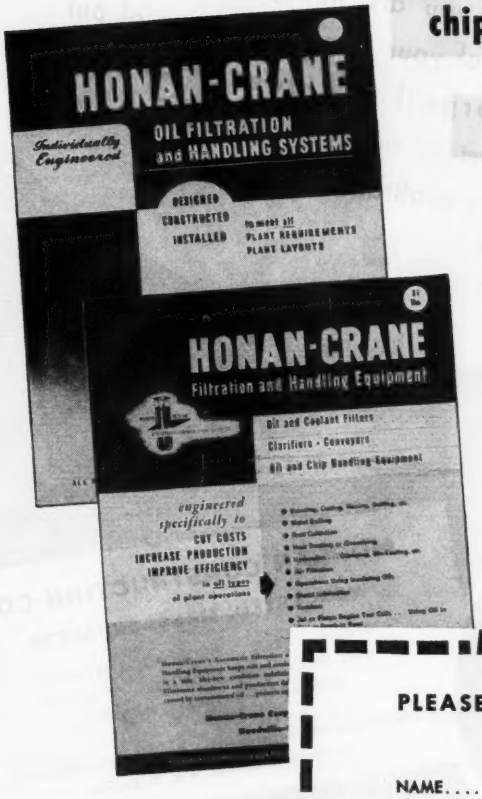

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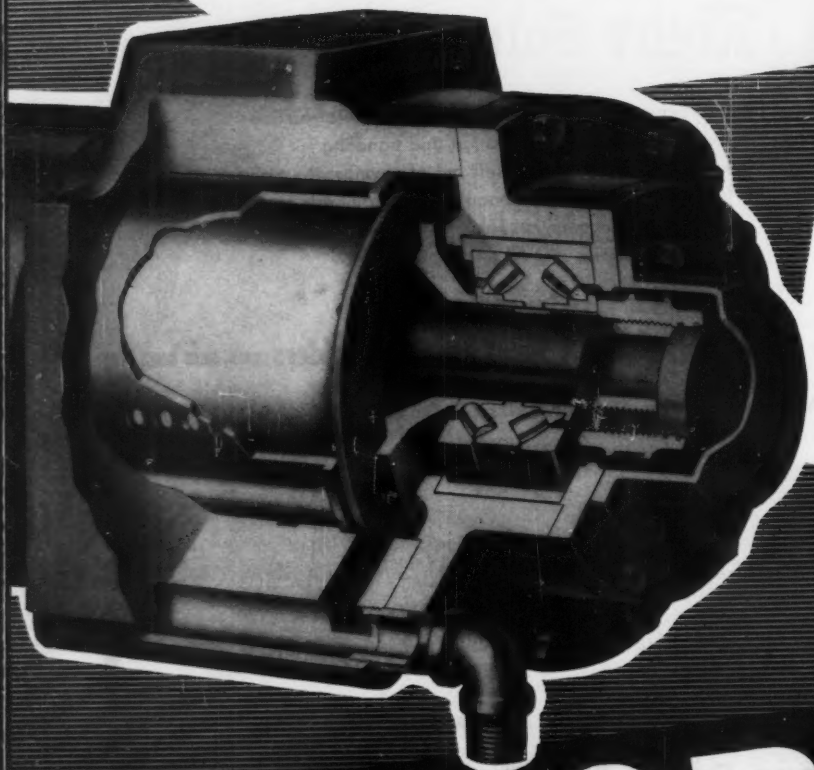

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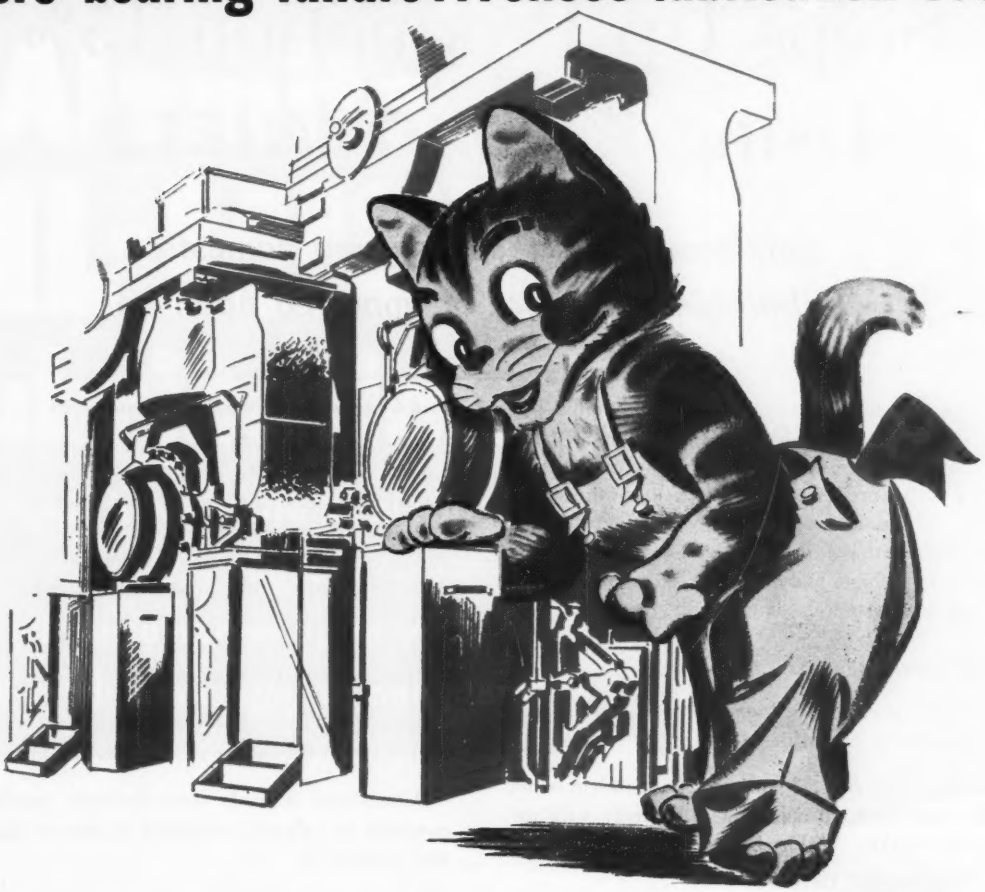

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Major Tin Plate Producer Steps Up Production 15% Using **PALMOSHIELD**

Sets production records using domestic Palmoshield in place of imported palm oil

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Actual mill experience in dozens of plants has proven these advantages of Palmoshield:

To Purchasing Agents. The price of Palmoshield is not artificially controlled. It rises and falls freely with the domestic fat market. Within the past year users have received several voluntary price cuts.

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Palmoshield requires absolutely no changes in rolling mill operation.

Palmoshield makes steel quicker to gauge and shape.

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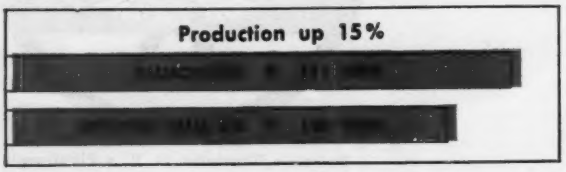

Quick Acceptance by steel producers

Nearly two-thirds of the major tin plate mills in this country and Canada are now using Palmoshield either in regular mill operations or for on-the-job-test runs. One company using Palmoshield for the first time was only 10 tons shy of the daily record. Another broke the single-shift record. And still another broke both 8-hour and 24-hour records.

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Palmoshield is subject to exact chemical control. Free fatty acid content can be controlled to within ½%.

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cuts oil consumption up to 90%

boosted production 50% at one plant!

Here from Alemite—world leader in lubrication—is lubrication progress so major—so far reaching—as to command the interest of *every* executive concerned with industrial lubrication practices and costs. Alemite Oil-Mist! The most efficient, continuous, fully automatic system in the field of machinery lubrication.

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- Air pressure seals bearings against dirt.
- Reduces oil consumption—up to 90%.
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1953 ASLE Annual Meeting

April 13-14-15

Hotel Statler Boston

PROGRAM PLANS for the 8th Annual Meeting and Exhibit of the American Society of Lubrication Engineers to be held on April 13, 14, and 15, 1953, at the Hotel Statler, Boston, include three symposiums, a round table forum, and thirty-four technical papers. A portion of this program includes the following:

Symposia: DIAGNOSING BALL AND ROLLER BEARING TROUBLES BY VISUAL INSPECTION
DIAGNOSING SLEEVE BEARING TROUBLES BY VISUAL INSPECTION

Forum: ASPECTS OF HYDRAULIC MACHINERY AND EQUIPMENT (with audience participation)

Papers: DESIGN AND SERVICE REQUIREMENTS DICTATE THE CHARACTERS OF OIL ADDITIVES, by E. C. Knowles & G. R. Furman, The Texas Co.

PROPERTIES OF LUBRICANTS THAT ARE DESIRABLE FOR USE WITH VARIOUS TYPES OF BEARING METALS

VISCOSITY STUDIES OF FLUID LUBRICATION AT HIGH PRESSURES, by Dr. L. B. Sargent, Jr., Aluminum Research Laboratory

AIR BEARING STUDIES AT NORMAL AND ELEVATED TEMPERATURES, by J. D. Pigott & E. F. Macks, N.A.C.A.

DEVELOPMENT OF SIGNIFICANT BENCH TESTS FOR AIRCRAFT TURBINE OILS, by A. E. Ryder, Pratt & Whitney

LUBRICATION IN THE DESIGN AND OPERATION OF TEXTILE MACHINERY, by R. M. Jones, Saco-Lowell Shops

Lube Lines

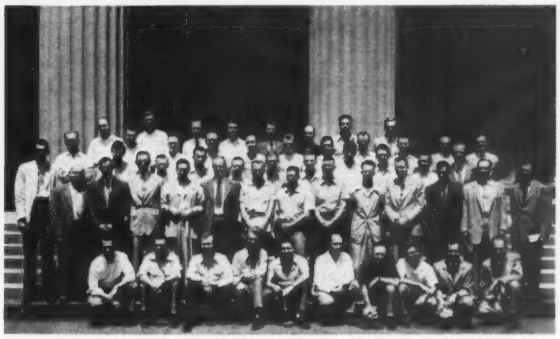
SEASON'S GREETINGS: The Officers and Directors of The American Society of Lubrication Engineers, as well as the Staff of *LUBRICATION ENGINEERING*, take this opportunity to wish you all a Merry Christmas and a happy and prosperous '53.

SYMPOSIUM HI-LITES: Two hundred and fifty-seven persons attended the First National Symposium on the Fundamentals of Friction and Lubrication in Engineering (the Victor A. Ryan Memorial Symposium) held in Chicago on September 8 & 9, 1952. Dr. M. E. Merchant, The Cincinnati Milling Machine Co., ASLE President, is shown above presenting the welcoming address. Also pictured are three of the speakers (from left to right): Prof. D. D. Fuller, Columbia University; Dr. A. Bondi, Shell Development Co.; and J. Boyd, Westinghouse Electric Co. A proceedings of the meeting will be made available, the time and price to be announced later.

SYMPOSIUM ON BEARING FATIGUE: Bearing failure is a problem that has confronted engineers ever since the inception of the first reciprocating engines. In the early days, fatigue was usually combatted by the simple experiment of making bearings large enough so that the unit loading would be low. With the development of the internal combustion engine came the necessity of raising bearing loads and at the same time reducing bearing size. This requirement called for drastic changes in both design and materials. With the passage of time, continued improvements along these two lines have resulted in performance, which by former standards, are truly remarkable. In the *Symposium On Bearing Fatigue* in this issue, which was sponsored by the Technical Committee on Bearings and Bearing Lubrication, the authors present various aspects of the current thought on the fatigue problem. Both Mr. Etchells and Mr. Palsuich discuss the matter of design and materials. The influence of the lubricant is considered by Messrs. Carmichael and Purdy.

J. Boyd

PLANT MAINTENANCE CONFERENCE PREVIEW: Sixty-six sessions on maintenance problems of the automotive, chemical, electrical manufacturing, food processing and packaging, foundry, paper mills and paper products, petroleum refining, printing and binding, rubber goods, steel mills and fabricating plants, and textile industries have been scheduled for the Plant Maintenance Conference to be held at the Public Auditorium, Cleveland, January 19 through 22, 1953, in the form of three panels, sixteen sectional conferences, forty-two roundtables, and four plant tours. Plant engineering problems will be discussed in separate groups, divided according to the number of maintenance employees engaged in the work. Panel discussions, involving top management policy, will include: *Planning and Scheduling Maintenance Work, Maintenance Costs and Budgets, Training Maintenance Workers and Supervisors, Work Standards and Measurements,* and *Growing Pains of the Engineering-Maintenance Organization.* Other topics to which conferences will be devoted include organization, operating policies, preventive maintenance, building maintenance, mechanical equipment, electrical equipment, power plant equipment and prevention and control of corrosion. Advance registration cards and hotel information may be obtained from Clapp & Poliak, Inc., 341 Madison Ave., N. Y. 17, N. Y.



M. I. T. LUBRICATION ENGINEERNG COURSE HI-LITES: Forty-seven students attended the Lubrication Engineering Course at the Massachusetts Institute of Technology in Cambridge, June 23 to July 3, 1952. Lecturers included: B. G. Rightmire, M.I.T.; A. F. Underwood, Research Laboratories Div., G.M.C.; E. R. Booser, General Electric Co.; E. Rabinowicz, M.I.T.; I-Ming Feng, M.I.T.; D. T. Wilcock, General Electric Co.; A. Bondi, Shell Development Co.; W. H. Millet, Carbide & Carbon Chemicals Co.; C. C. Currie, Dow Corning Corp.; D. M. Cleaveland, Bendix Aviation Corp.; W. E. Campbell, Bell Telephone Laboratories; M. C. Shaw, M.I.T.; A. R. Jones, Fafnir Bearing Co.; J. H. Fuller, E. I. du Pont de Nemours & Co.; R. J. Torrens, Eastman Kodak Co. The Class Roster (from left to right): 1st row: H. F. Szepan, C. D. Gilmore, R. E. Prendergast, K. Campbell, J. S. Acterman, J. A. Calleo, E. Y. Soomil, F. Heymann, E. Rabinowicz, D. L. Sheller; 2nd row: A. B. Jones, E. R. Keast, N. J. Ireland, J. R. Hicks, M. Z. Fainman, R. L. Brady, W. W. Holt, V. L. Adams, L. B. Ward, G. H. Vorhoff, T. F. Hand; B. T. Schleicher, S. L. Stalson, A. Bondi; 3rd row: M. O. Dion, T. W. Havely, L. W. Weber, J. A. Travis, G. C. Ambler, J. J. O'Donnell, E. Holodnik, H. H. Rowand, L. V. Collings, G. T. O'Brien, E. C. Helmke; 4th row: W. F. Dalheim, A Lewis, H. H. Christen, C. C. Raymond, H. R. Taylor, L. D. Krapf, W. H. Pfeiffer, S. H. Rice, A. Crankshaw, W. E. Campbell; 5th row: A. G. Almack, N. W. Gits, E. L. Anderson, J. N. Williams, B. G. Rightmire, W. P. Youngclaus Jr., R. E. Speers, and A. J. Romeril.

VOLUME VI, Proceedings of the National Conference on Industrial Hydraulics, containing the twenty technical papers presented at the 8th Annual Meeting in Chicago, September 4 & 5, is available at $4.50 per volume. A limited supply of Proceedings of former meetings is still available. For

(Continued on page 306)

PLANNED LUBRICATION AS A PART OF PLANT MAINTENANCE*

by T. R. Witt
Tennessee Eastman Co.
Div. of Eastman Kodak Co.
Kingsport, Tennessee

Mr. Witt has been in mechanical work since 1917. Employed by the Tennessee Eastman Co. in 1930, he has held the positions of Foreman, Genueral Foreman, Supervisor and the present position of Division Maintenance Engineer of the yarn divisions. He is a member of ASLE, ASME, and The Tennessee Society of Professional Engineers.

ABSTRACT: There was a time when lubrication was left more or less to the "oiler." For the past several years lubrication has been made a specialized field of maintenance and is now controlled from the purchase of the lubricant to the final application. The lubrication of the equipment is the responsibility of the division maintenance department and is performed by a group of lubrication men under the supervision of a lubrication foreman. This foreman is also responsible for preparing schedules, keeping all necessary records and training lubrication men. This is a more effective lubrication program and as a result lubrication costs have been sharply reduced.

The maintenance of a modern industrial plant is big business and the lubrication of the equipment is by no means an insignificant part of that business. It is not necessary to belabor the importance of lubrication, because it is well known that machinery will not run very long without grease and oil. It must be assumed, therefore, that all of the machinery that is in operation is, in some manner, being lubricated. However, in many cases this vital part of plant maintenance probably does not receive the attention that is given to it by some plants and the lubrication program is, at best, an uncertain one. This lack of interest in lubrication can result only in the failure to take advantage of all of the improvements that have been made in lubricants within the past few years, as well as all of the valuable information that has been and continues to be published on the subject.

To take advantage of the research and development which the manufacturers of lubricants and lubricant appliances have put into their products requires an organization composed of specialists in the various phases of lubrication. An organization of this kind, when properly established, is in a good position to supervise the lubrication program from the purchase of the lubricant to the final application and thereafter the schedules of relubrication. The mechanics of such a set-up will, of course, vary widely among industrial plants, and it is the purpose of this discussion only to outline briefly how the lubrication program has been improved, over a period of years, in one large plant.

Several years ago our management set up an organization to investigate plant lubrication and to take required steps to do the best possible lubrication job. This organization is composed of the maintenance engineers from the various divisions, with one member from each of the staff organizations of Purchasing, Stores, Industrial Laboratory, Materials Standards, and Engineering Departments and is known as the Lubrication Committee.

*Paper presented at 7th Annual Meeting, ASLE, Cleveland, April 7, 1952.

The committee meetings were held once each month until most of the ground work was done, after which the meetings were scheduled quarterly. A great deal of benefit has been derived from these meetings by the exchange of information and experiences regarding lubrication problems all over the plant. Also, close liaison is maintained with the lubrication people of other divisions of the Kodak Company.

Some of the accomplishments of this group have been the standardization of lubricants; the development of lubricant specifications; laboratory tests to see that all lubricants meet specifications; distribution of information throughout the plant by meetings, conferences and publication of a lubrication practice manual; approval of equipment design to provide for adequate lubrication; and many others.

The first thing to be considered in reviewing such a program is the type and number of lubricants required to do the job adequately. Lubricating a plant with the fewest number of lubricants practicable greatly reduces the problems of selection, purchasing, storage and application. In the beginning it was found that 57 types of greases and oils were carried in stock and considered necessary for the lubrication of the plant. After a thorough investigation and after much discussion, the number was reduce from 57 to 12 standard types. The characteristics of the lubricants that are purchased on an annual contract basis are described in Table I.

A specification is written for each lubricant that is carried in stock. The specifications describe the limits in viscosity, flash point, pour point, rust and oxidation additives and other characteristics required, and also the laboratory methods to be used in checking these characteristics. All specifications are numbered and the vendors are requested to mark all containers with this number. It is also the stock number. The oils and greases are requisitioned and issued by this number, and it is used in all schedules and records that are made up and kept by the maintenance departments. No trade names are used.

The Purchasing Department does not buy lubricants for general plant use except on a specification basis and with the approval of the lubrication committee. Laboratory tests are made and reported by the specification number to control the quality of the incoming lubricants. Each shipment must be approved by the laboratory before it is released for use in the plant.

The Engineering Department representative on the committee has the responsibility of seeing that all drawings

for new equipment are checked while in the design stage to ensure that lubrication is provided for in the design. This is preventive maintenance put into effect even though the machine is still on the drawing board. Machine designers are usually troubled by a good many problems, and effective lubrication might not be considered as it should be unless given this special attention.

LUBRICATION SCHEDULE - TWISTING DEPARTMENT - BUILDINGS 59 and 70 MODEL "C" TWISTERS

Machine Part	Lubricant	Frequency	Method	Remarks
Hand end gearing	Oil - THC 2007	Every week	Flow'o oil can	Fill oil cups and oil gears
Silent chain	Oil - THC 2007	Every day	Flow'o oil can	2 shots when machine is in operation
Jack shaft	Grease - THC 2027	Every 2 years	Hand gun	3 shots
Feed roll comp. gears (Ball bearing)	Grease - THC 2027	Every 2 years	Hand gun	3 shots
Feed roll gears (Ball bearing)	Grease - THC 2027	Every 2 years	Hand gun	3 shots
Motion rail drive gears	Oil - THC 2007	Every week	Flow'o oil can	A few drops on gears
Foot and oil rail gears	Oil - THC 2007	Every month	Flow'o oil can	Keep reservoir to proper level
Twist gear shaft reservoir	Oil - THC 2007	Every month	Flow'o oil can	Replenish waste and fill reservoir
Winddown gear shaft reservoir	Oil - THC 2007	Every week	Flow'o oil can	Replenish waste and fill reservoir
Cam gear reservoirs	Oil - THC 2007	Every month	Flow'o oil can	Keep reservoir nearly full
Ring rail and thread board lift rods	Oil - THC 2007	Every month	Flow'o oil can	Two or three drops to each
Pull bar pins and cams	Oil - THC 2007	Every month	Flow'o oil can	Two or three drops to each
Ring rail and thread board lift rod foot rollers	Oil - THC 2007	Every 6 months	Flow'o oil can	A few drops to each bearing
Foot and tape drum bearing	Grease - THC 2028	Every 2 years	Hand gun	Spot check before adding grease
H. E. and Intermediate tape drum bearings	Grease - THC 2027	Every 2 years	Hand gun	Spot check before adding grease
Cross shaft bearings (Open bearings)	Oil - THC 2007	Every 2 months	Flow'o oil can	Two or three drops to each
Cross shaft bearings (Plain bearings)	Oil - THC 2007	Every 2 months	Flow'o oil can	Two or three drops to each
Cam shaft bearings	Oil - THC 2007	Every month	Flow'o oil can	Fill oil cups
Cam roller	Oil - THC 2007	Every month	Flow'o oil can	Fill oil cups
Builder chain	Oil - THC 2007	Every month	Flow'o oil can	A few drops on chain
Chain pulley and carriage	Oil - THC 2007	Every month	Flow'o oil can	Fill oil cups and oil holes
Builder arm bearing	Oil - THC 2007	Every month	Flow'o oil can	A few drops in oil holes
Yardage Counter gear felts	Oil - THC 2007	Every 2 weeks	Flow'o oil can	One or two drops on felts
Yardage counters	Oil - THC 2002	Every month	Flow'o oil can	One or two drops in cups

Fig. 1 A sheet of the lubrication master schedule describing the department, location of the equipment by building, and the particular type of equipment. Machine parts are listed by name, the type of lubrication to be used, the frequency, and the method of application, with special instructions noted in the last column.

The scheduling, application, and general administration of the lubrication of the equipment in the various divisions is conducted by the division maintenance departments. Although the other divisions have similar procedures, the following discussion applies mainly to the two divisions that manufacture cellulose acetate yarns. The equipment, which is both chemical and textile equipment, is in continuous operation and consists essentially of conveyors, dissolvers, blenders, pumps of many types and sizes, filtration equipment, spinning, twisting, and winding machinery.

After a thorough investigation of the requirements of the job, it was decided that a maintenance foreman does not have the time, the interest, nor the necessary knowhow to supervise the lubrication of the machinery in his area. The schedules cannot be set up and then forgotten, because there are many changes that have to be made as the program progresses. There are new lubricants to be evaluated and adopted changes in lubrication schedules to conform to machine alterations, new machinery to be scheduled, lubrication men to be trained, etc. All this requires full time supervision and the supervisor must know something about the fundamentals of lubrication. Most maintenance foremen are not interested enough in lubrication to read anything about it nor to give much thought to it other than casual observation.

Lubrication has been set up as a specialized field of maintenance, and we have organized a group of men, "lubrication men", under the supervision of a "lubrication foreman". This group has no responsibilities other than lubrication, from the cleanliness of the oil storage rooms to the final application. These men are carefully instructed in all phases of their work. They are given talks on the points of lubrication that we feel they are capable of understanding, which includes the simple mechanics of lubrication and the principles of the machines they lubricate. It is pointed out to them that they are in a good position to know of incipient failures and to help measure the effectiveness of the lubricants. They are instructed to report to the lubrication foreman any unusual conditions found in the machinery, or any part that for any reason cannot be lubricated properly. They are furnished all the necessary tools and equipment with individual lockers to keep them in, and they are required to see that they are in good condition. A group of lockers is shown in Fig. 4. Storage rooms have been conveniently located, and periodic inspections are made to keep them clean and in order. The importance of keeping lubricants free of dirt and grit is repeatedly pointed out to the men. The emphasis put on lubrication, the attention and supervision the lubrication men have received, have created job interest and given them a feeling that their job is important and not one to be done haphazardly. The pay rate for this classification is between the mechanic second class and mechanic helper brackets.

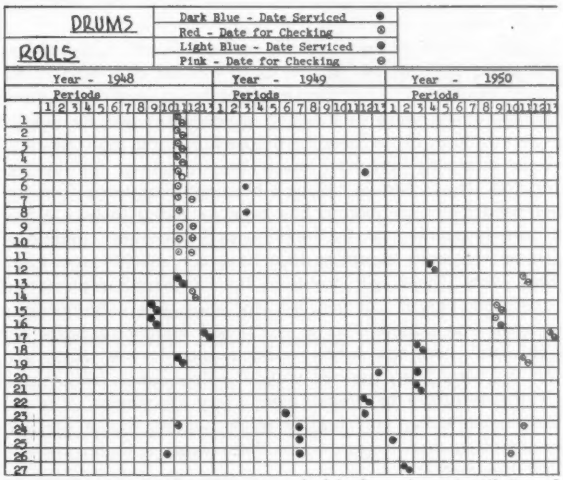


Fig. 2 A portion of the lubrication schedule for a large installation of ball bearings using colored map tacks to indicate the date lubricated and the date to be checked.

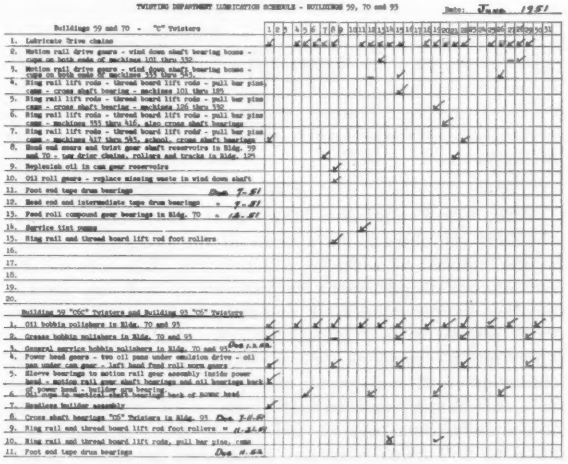

TWISTING DEPARTMENT LUBRICATION SCHEDULE - BUILDINGS 59, 70 and 93
Buildings 59 and 70 - "C" Twisters
Building 59 "C&C" Twisters and Building 93 "C&" Twisters

Fig. 3 A portion of a monthly lubrication schedule listing all equipment one man is to service or be responsible for, with all parts to be serviced listed on the left side of the sheet and the scheduling of the work on the right half under the days of the month. As each job is finished, its completion is indicated with a check mark. Should the lubrication man be unable to complete the job on the day started, he makes a cross mark as in item 10 on the bottom half of this schedule, reports to his foreman what part of the work remains to be done, and on the date of completion indicates with the check mark in the proper date square on the schedule.

The frequency of lubrication is set up on the recommendation of a competent "lubrication engineer". We rely on his experience as to how long a certain bearing should run without greasing. When a lubrication schedule has been set up for a particular machine, it is supervised closely and inspected frequently until the requirements have been established, and the schedule is then modified accordingly. The best lubrication engineer will occasionally misjudge just how long a lubricant will last. The follow-up is the re-

sponsibility of the maintenance group and it requires a lot of close observation and supervision to keep up with it.

A few representative types of the forms that are used to schedule and record the frequency of lubrication are partially shown in Fig. 1, 2 and 3. The forms are designed so all information regarding the lubrication of all of the equipment is clearly and continuously available to all people concerned with this phase of plant maintenance. They also minimize clerical work and the keeping of records and permit a quick follow-up of the work by supervision.

An attempt is made by this method of scheduling to use an adequate amount of lubricant without over-lubricating. Over-lubrication is a waste of material and labor and generates excessive heat which, if continued, will eventually cause failure. In the case of grease-lubricated ball bearings it is surprising just how much the frequency can be extended. The schedule for ball bearings that were formerly greased every six months has been extended to two to five years, depending on the location and operating conditions. There are many bearings that have run much longer than five years. There is a group of 10 H.P. electric motors that operated almost continuously for nine years before any grease was added. This is mentioned merely to point out that ball bearings, when good lubricants are used, will run for very long periods without relubrication and that it is not necessary to grease them every six months. It is our opinion now that any ball or roller bearing that is greased more often than every two or three years is being greased too frequently.

TABLE I — CHARACTERISTICS OF LUBRICANTS

OILS

	Lubricant No.	Flash F. Minim.	Viscosity 100 F.	Viscosity 210 F.	Pour Point F. Max.	Acid No.	Rusting and Oxidation Stability	Saponifiable Matter
Light Mineral Oil	2000	275	70/80		25	0.1	-	-
Turbine Oil	2001	360	140/160		20	0.1	Pass Test	-
Heavy Lubricating Oil	2002	475	600/700	(40 Min.)	20	0.1	Pass Test	-
Steam Cylinder Oil	2003	450		140/160	45	0.2	-	5/7%
Straight Mineral Oil	2004	350	3500/4000	170/185	45	0.1	-	-
Medium Lubricating Oil	2005	410	275/375	50 (Min.)	15	0.1	Pass Test	-
Ammonia Compressor Oil	2006	350	200/300	45 (Min.)	-20	0.1	-	-
Extra Heavy Lubricating Oil	2007	450		90/120	40	-	-	-
Heavy Engine Oil	2008	500		118/160	40	0.1	-	-

GREASES

Grease Base		Viscosity of Oil at 100 F.	Minimum Melting Point F.	Penetration Worked	Filler
2025	Calcium	250/300	295	265/295	None
2027	Calcium & Sodium	485/535	300	220/250	None
2028	Sodium or mixed	275/300	390	275/325	None

After good lubricants have been provided and the schedules established, the success of the program depends on the manner in which it is supervised. This part of the Maintenance Department's responsibilities will lag behind all other maintenance work unless the supervision takes an active interest in it. It cannot be fully effective unless it is supervised by someone who is interested and understands lubrication well enough to appreciate the importance of the small details, such as cleanliness of oil storage rooms, keeping lubricants covered and free of dirt, etc. People are prone to drift away from any established procedure unless their activities are followed closely.

The lubrication foreman trains and supervises the lubrication men. He gives personal attention to the training of new people on the job. He checks each man often to see that he has not fallen into bad practices of any kind, and he frequently checks each man in the field on the kind of lubricant he is using. The foreman formulates the schedules, the frequency being determined with the help of maintenance supervision; keeps all necessary records and keeps an adequate supply of lubricants in all oil rooms; checks all cases where there is any question as to whether the frequency or the lubricant is adequate. He periodically checks all oil rooms to see that they are clean and in order; checks the lockers, with each man, to eliminate the accumulation of unnecessary junk; and checks each man's equipment to assure that he has what is needed to do the job and that it is in good condition.

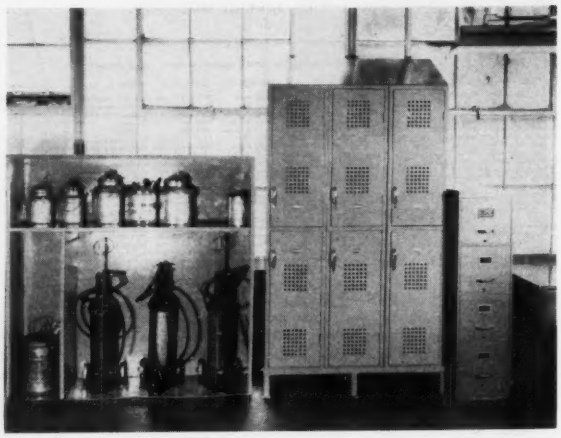


Fig. 4 Lockers in one of the oil rooms which are provided for the lubrication men to help build morale and incite more pride in the appearance of the oil rooms and the job in general.

Many problems were encountered in setting up these procedures and many old practices were difficult to change. When men have worked for a long period of time with little supervision and very few instructions, doing the job more or less their own way, changing their attitudes is difficult. It is easy enough to get lubrication men to agree that too much grease will cause a ball bearing to heat, but it is hard to keep them from pumping a housing full. They usually feel that too much is better than not enough and if kept full they cannot be accused of missing a bearing. It was a little difficult for them to realize that the old methods were being changed and that Supervision intended to follow the program closely to see that lubrication was carried on in accordance with the new set-up.

The advisability of purchasing the more expensive lubricants, especially the ball bearing greases that had been recommended, came up for a lot of discussion. Everyone agreed that if the grease was applied by the usual practices much would be forced through the housing and more wasted than the amount necessary to lubricate the bearings. It was agreed also that unless the proper amount was used in bearing housings the purpose of the better grease would be defeated and a cheap grease would probably do as well. This had to be overcome, of course, by establishing the necessary frequencies and methods of lubrication and supervising and training the lubrication men.

In order to keep machinery in operation there are certain preventive measures that should be taken when the machine is assembled, and at regular intervals while it is in operation. The first step of course is good mechanical work. A study of the basic theory of lubrication brings out very clearly the importance of good mechanical work. It is claimed for instance that 90% of all ball bearing failures are caused by dirt. When it is realized by maintenance

(Continued on page 306)

CLEANING LUBRICATION SYSTEMS, by A. S. Morrow, Shell Oil Co., will be featured in the February issue of *LUBRICATION ENGINEERING* as the fifth in the series of Practical Lubrication articles. At the conclusion of this series, the Society will publish the papers in book form as another of its practical lubrication manuals.

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LUBRICATING PROPERTIES OF MOLYBDENUM DISULFIDE*

by I-Ming Feng
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Dr. Feng received his B.S. degree in Mechanical Engineering from Chiao-Tung University, China, in 1946; he received his M.S. degree in 1948 and his Ph.D. degree in 1952 from the University of Michigan. Since 1950 he has been engaged in research on solid lubricants at the Lubrication Laboratory of the Massachusetts Institute of Technology.

ABSTRACT: Research results regarding the lubricating properties of molybdenum disulfide are reported in this paper. Its crystal structure, the methods of applying the compound, its friction coefficient under ordinary conditions, the result of X-ray diffraction study, and the effects of various factors, including the surrounding atmosphere, the temperature, the sliding speed, the normal load, and the crystallographic orientation are discussed. Molybdenum disulfide is also frequently compared with graphite.

INTRODUCTION: Research work in studying the lubricating characteristics of solid lubricants is becoming more active every day. This is the result of an ever increasing number of special problems that call for the use of dry lubricants, both in industry and laboratory applications. Among the solid lubricants, molybdenum disulfide MoS_2 has received an increasing amount of attention, because it offers a solution to many problems which cannot be solved by using ordinary lubricants. One example is the lubrication of the bearing inside a vacuum chamber where an ordinary lubricant cannot be used and graphite fails to lubricate. Graphite is a well known solid lubricant very similar to molybdenum disulfide, that is frequently discussed and compared with molybdenum disulfide.

CRYSTAL STRUCTURE OF MOLYBDENUM DISULFIDE: The use of molybdenum disulfide as a solid lubricant is suggested by its laminar structure, which is similar to that of graphite. The crystal structure of graphite consists of laminae of linked hexagons of carbon atoms as shown in Fig. 1a. Each carbon atom is surrounded by four neighboring carbon atoms. Three of the four neighbors lie in the same lamina at a distance of 1.42 Å from the central atom, and the fourth one is at a much greater distance of 3.40 Å. In other words, the forces holding the atoms in a lamina together are much stronger than those between neighboring laminae.

Fig. 1b shows the crystal structure of molybdenum disulfide. The structure is also laminar in nature but, in this case, each lamina is a sandwich made up of two layers of sulfur atoms with a layer of molybdenum atoms in between. Within each of the layers, the atomic arrangement is hexagonal. All laminae are identical in having this same sandwich construction. The force holding the atoms in each lamina together are stronger than those attracting the sulfur atoms in one lamina to those of another.

An electron micrograph of a molybdenum disulfide flake is shown in Fig. 2a. The magnification of this micrograph is indicated by the scale on the picture. The specimen was prepared by placing a drop of distilled water containing dispersed molybdenum disulfide powder on a thin collodion film which was supported on a screen, and then allowing the water to dry. Examination of the specimen with the aid of electron microscope tells that almost all molybdenum disulfide flakes were lying flat on the collodion film (perpendicular to the electron beam). The one which is shown in Fig. 2a happened to stand up with its cleavage plane parallel to the electron beam and thus reveals its laminar structure. The interpretation of this electron micrograph is given in Fig. 2b. It is to be mentioned that each of the plates shown in Fig. 2b is actually consisting of a great number of laminae.

The similarity in crystal structure of these two substances is rather striking. Graphite is well known as a good solid lubricant. Its lubricating property has been attributed to its particular crystal structure, that is, to the fact that the weak bonds between graphite laminae give an easy sliding of one lamina over another. Recent experimental results[1,15,16] indicate that the lubrication of graphite is not intrinsic in its laminar crystal structure alone, but also depends upon adsorption films on the laminae faces. From the similarity in crystal structures, molybdenum disulfide is naturally expected to have a good lubricating property. Strangely enough, the lubricating characteristic of molybdenum disulfide has been found to be independent of adsorption films[2]. Because of this difference, molybdenum disulfide surpasses graphite as a solid lubricant under conditions where adsorption films cannot form on the cleavage faces.

METHODS OF APPLYING MOLYBDENUM DISULFIDE: Molybdenum disulfide powder can be rubbed onto the surface of the base metal to form a thin film without the aid of any binding materials. This can be done by rubbing the surface with molybdenum disulfide powder with the fingers or a pad of cotton. The powder can also be compressed into a cake which is then pressed against a moving surface to form a thin film on the latter. Sumner[3], Bowden[4], and Bell and Findlay[5] have reported that the rubbed-on film adheres to metal reasonably well. Stuart[6] has reported that, in contrast to a graphite film, the molybdenum disulfide film on a metallic surface is held with exceptionally good adhesion. According to the author's own experience, a rubbed-on molybdenum disulfide film adheres to metal better than does a graphite film, which observation is thus in good agreement with that of others. The reason why it adheres so well to metallic surfaces has not yet been definitely established. According to Bell and Findlay[5], the adherence between a molybdenum disulfide lamina and a metallic surface is due to a strong metal to sulfur bond.

Molybdenum disulfide surface films can also be produced by spraying or brushing a dispersion of the powder in a volatile liquid medium onto a surface and allowing it to dry.

The method of using a binding material to produce a molybdenum disulfide film on metallic surface has also been used by various investigators. Johnson et al[7] have used corn syrup with success to produce a tenacious film. The detailed method of preparation is given in NACA TN

* Sponsored by the Technical Committee on Lubrication Fundamentals and presented at the 7th Annual Meeting, ASLE, Cleveland, April 9, 1952.

No. 1578. Bowden and Shooter[8] have used ebonite.

A molybdenum disulfide film may also be prepared by changing chemically the surface layer of metallic molybdenum or its alloys into molybdenum disulfide. Bowden[4] has tried this method on porous sintered molybdenum. Of course, this method can only be applied to molybdenum or its alloys.

Milne[9] has used the method that involves phosphating the metallic surface to produce a non-metallic surface layer having a porous structure which can then be filled with molybdenum disulfide powder.

The method of impregnating the base material with molybdenum disulfide powder has been used by Bowden and Shooter[8], and also by Johnson, Swikert and Bisson[7]. Porous sintered copper, porous sintered silver and perspex have been tried. In all cases, the friction was reduced due to the presence of molybdenum disulfide. By this means, the molybdenum disulfide can be incorporated into almost any material.

FRICTION COEFFICIENT OF MOLYBDENUM DISULFIDE UNDER ORDINARY CONDITIONS: Before the discussion of the effect of various factors upon the lubricating characteristic of molybdenum disulfide, we shall discuss the friction coefficient of molybdenum disulfide under ordinary conditions, that is, with a reasonable speed, under a moderate load, at room temperature, and in the ordinary atmosphere.

It would be helpful for the following discussion to mention that the major component of friction is the shear component and the friction coefficient is approximately equal to the ratio of the shear stress s and the flow pressure p (Cf. references [10,11,12]). In the case of rubbing two surfaces which are covered by thin molybdenum disulfide films, shear will occur inside the film, i.e. s is determined by the shear strength of the molybdenum disulfide, whereas the actual area of contact is determined by the hardness of the base material. By changing the hardness of the base material, one can change the flow pressure p. Thus, the friction coefficient can be lowered by increasing the hardness of the base material. This has been found to be so by the author.

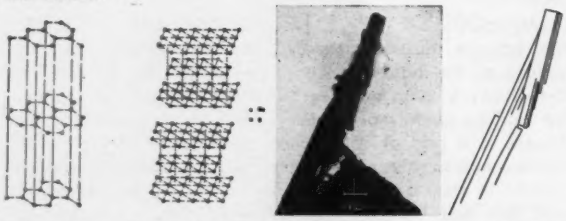

Fig. 1a & 1b (left) Crystal structure of graphite, and molybdenum disulfide, respectively.

Fig. 2a & 2b (right) Electron micrograph of a molybdenum disulfide flake which stands up with its cleavage plane parallel to the electron beam and demonstrates its laminar structure, and interpretation of the electron micrograph, respectively.

Table I gives the values of friction coefficient of molybdenum disulfide reported by various investigators. The first four data (a), (b), (c), and (d) are representative values of friction coefficient because the method of applying molybdenum disulfide or graphite to the surface by rubbing introduces no foreign material. Furthermore, the experimental conditions are roughly the same, and hence the data are directly comparable. The values of friction coefficient for molybdenum disulfide obtained by various investigators agree fairly well and are in the range from 0.05 to 0.095. The range of friction coefficients for graphite is from 0.11 to 0.19. This indicates that the friction coefficient of a thin film of graphite on a metallic surface is higher than that of a thin molybdenum disulfide film on the same metallic surface.

The friction coefficient of a molybdenum disulfide film prepared with the aid of a binding material [Table I, (e) and (f)] is of the order of 0.20 which is higher than that of rubbed-on films which give a value ranging from 0.05 to 0.095. The explanation for this is that the film prepared with the aid of the binding material is usually much thicker than the rubbed-on film, and hence the hard backing material does not help in reducing the friction coefficient. This explanation is further supported by the fact that the friction coefficient of the film prepared in this way is of the same order of magnitude as that of a compressed molybdenum disulfide pellet [Table I, (i)].

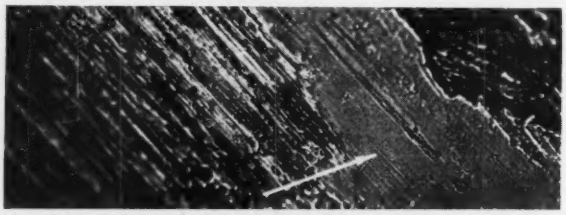

Fig. 3 Photomicrograph showing the retained molybdenum disulfide film (indicated by arrows) on the steel rider (500X).

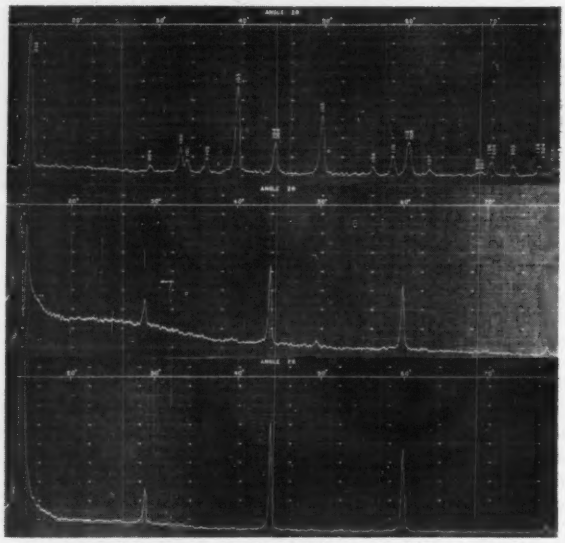

Fig. 4 X-ray diffraction results of molybdenum dissulfide. 4a (top) Randomly oriented molybdenum disulfide powder sample. 4b (center) Molybdenum disulfide surface film which was rubbed on the ground glass plate of 30 microinches r.m.s. 4c (bottom) Molybdenum disulfide surface film on the 30 microinches ground glass plate after it has been rubbed over about ten times by the glass slider under a load of 3,000 grams.

EFFECT OF ATMOSPHERE: It has definitely been established that graphite loses its lubricating property in the absence of condensable vapor[1,15,16]. In vacuum, wear of graphite accompanied by high friction is very rapid. Suitable additives to graphite have been found which compensates for its tendency to lose lubricating capacity at high altitudes where semi-vacuum conditions prevail. However, molybdenum disulfide behaves differently. Savage's experimental result[2] shows that the wear of molybdenum disulfide in vacuum (0.05 micron) is negligibly small. This indicates that it is suitable for lubricating moving parts in vacuum. As a matter of fact, it has been applied to the rotating target X-ray tube and found to work extremely well.

EFFECT OF TEMPERATURE: Because the lubricating property of molybdenum disulfide is independent of adsorption films, this compound is expected to be a good

lubricant at low temperatures, as well as high temperatures. However, at high temperatures, its stability and oxidation characteristics have to be considered. In an inert atmosphere, molybdenum disulfide is extremely stable at high temperatures. The Handbook of Chemistry and Physics[17] indicates the melting temperature of this compound to be 1185 C. (2160 F.). However, no melting was observed on heating molybdenum disulfide in argon up to 1427 C. (2600 F.)[18]. Nevertheless, it decomposes at a temperature around 1100 C. (2010 F.) in vacuum, yielding metallic molybdenum and free sulfur[18].

Godfrey and Nelson[19], and Ryozi[20] have studied the oxidation characteristics of molybdenum disulfide. Their results indicate no detectable oxidation at temperatures below 400 C. (750 F.). The products of oxidation of molybdenum disulfide formed above 400 C. do not have good lubricating property. The molybdenum oxide has strong abrasive properties and the sulfur is corrosive. Godfrey and Nelson[19] have shown that, at high temperatures, a molybdenum disulfide surface film maintains its low friction during its oxidation as long as an effective subfilm of molybdenum disulfide remains. Therefore, the use of molybdenum disulfide as a lubricant in an oxidizing atmosphere, at temperatures high enough to produce oxidation, should be carefully studied to be sure that an effective subfilm of molybdenum disulfide remains under operating conditions, or its use should be avoided.

METHOD OF APPLYING MoS_2 (or graphite)	INVESTIGATOR	DESCRIPTION OF THE EXPERIMENTAL CONDITIONS	FRICTION MoS_2	COEFFICIENT Graphite
Rubbing with MoS_2 (without binding material)	a. Peng	MoS_2 film rubbed on SAE 1020 steel speed 40 fpm. Load - 900 gm. (Friction coefficient of SAE 1020 steel under the same conditions - 0.55)	0.05-0.095	
	b. Peng	Graphite film rubbed on SAE 1020 steel Other conditions same as in (a).		0.11-0.14
	c. Bowden[4]	MoS_2 film rubbed on steel (friction coefficient of the steel - about 0.8)	0.05	
	d. Johnson, Godfrey & Bisson[3]	Graphite film rubbed on normalized SAE 1020 steel Speed about 50 fpm. Load - about 100 gm. (Friction coefficient of normalized SAE 1020 steel - 0.56)		0.19
With the aid of binding material	e. Peng	MoS_2 film prepared on SAE 1020 steel with the aid of corn syrup as the binding material. Other conditions same as in (a).	0.20	
	f. Johnson, Godfrey & Bisson[3]	MoS_2 film prepared on normalized SAE 1020 steel with corn syrup as the binding material. Other conditions same as in (d).	0.19	
By chemical means	g. Bowden[4]	MoS_2 formed chemically on sintered Mo (Friction coefficient of sintered Mo - 0.45)	0.1	
Porous phosphate method	h. Milne[9]	MoS_2 filled in the porous structure of the phosphate surface layer	0.04-0.1	
By compressing MoS_2 powder into pellets	i. Peng	MoS_2 pellets formed by compressing the powder in a pellet mold at a pressure of 50,000 psi. The compressed MoS_2 pellets were tested against discs of various materials, including brass, copper beryllium alloy, SAE 1020 steel, electroplated copper on SAE 1020 steel, various MoS_2 films on SAE 1020 steel, and also compressed MoS_2 itself.	0.30-0.22	
By impregnating the base material	j. Bowden[4]	MoS_2 impregnated in sintered copper (Friction coefficient of sintered copper - 0.3).	0.13	
	k. Johnson, Swikert & Bisson[13]	MoS_2 (35%) incorporated in hot-pressed bearing material of 60% Ag and 5% Cu (Friction coefficient of hot-pressed bearing material containing 95% Ag and 5% Cu - 0.30)	0.17	
	l. Bowden & Shooter[5]	MoS_2 (a few percent) in perspex (Friction coefficient of perspex - 0.45)	0.1-0.15	
By evaporating the liquid medium of a dispersion of the powder	m. Hughes & Whittingham[14]	Graphite film formed on mild steel surface from a colloidal graphite suspension. Speed - 0.06 fpm. Load range - from 50 to 100 g. (Friction coefficient of mild steel - 0.35)		0.09

Table I—Friction Coefficient of Molybdenum Disulfide.

Graphite does not melt either. Its sublimation temperature is above 3500 C. (6330 F.)[21,22]. According to Vosburgh[22], the temperature at which oxidation of graphite begins in air is 450 (842 F.). Because of the dependence of the lubricating property of graphite upon adsorption films, graphite is expected to function less satisfactorily as a lubricant at high temperatures than at room temperatures. Above the oxidation temperature, the oxidation of graphite produces no abrasive substance, and the carbon dioxide formed is generally non-corrosive. For these reasons, graphite is preferable at temperatures above the oxidation temperature of molybdenum disulfide.

EFFECT OF SLIDING SPEED: Because of the fact that graphite lubrication depends on the condensed vapor films and that the surface temperature increases with increasing sliding speed, graphite will not be expected to lubricate as well as molybdenum disulfide at high sliding speeds. This has been found to be so by Johnson, Godfrey, and Bisson[7]. Their experimental results also demonstrate that the friction coefficient of a molybdenum disulfide surface film decreases with an increasing sliding speed.

EFFECT OF NORMAL LOAD: Boyd and Robertson[23] have observed a decreased friction coefficient of molybdenum disulfide with increasing pressure. Milne[9] has reported that the friction coefficient of molybdenum disulfide is lower at heavy loads than at light loads. The experiments carried out by the author yielded similar results. In these experiments, a hard steel ball of 1/4" diameter was used as a rider sliding against a molybdenum disulfide film which was rubbed on the surface of a Cu-Be disc. The rider was sliding repeatedly over the same friction track on the Cu-Be disc at a load of 19.5 kg and a speed of 40 fpm, until the surface film was penetrated. Before the surface film was penetrated, the film which covered the friction track was shiny, smooth, and continuous. It was found that penetration usually started at some point which is presumably the weakest spot, and was followed by metal transfer which causes the sliding to become "rough" and a rapid spread of penetration. A retained surface film has been observed on the wear spot of the rider as shown in Fig. 3 and also inside the wear track of the disc. This indicates the tenacity of the molybdenum disulfide surface film, as well as its resistance to penetration.

EFFECT OF CRYSTALLOGRAPHIC ORIENTATION: The laminar structure of molybdenum disulfide suggests the dependence of its friction characteristics on its crystallographic orientation. Experiments were carried out to study this dependence. Specimens were cut from natural molybdenum disulfide crystals. With a freshly prepared cleavage surface of the crystal sliding against a SAE 1020 steel disc at a speed of 40 f.p.m. the friction coefficient was 0.10 in ordinary atmosphere. With the edges of the cleavage planes sliding against the same disc at the same speed, the highest value of friction coefficient observed was 0.26; and the actual friction coefficient is considered very possibly to be higher than 0.26, because, in preparing a surface perpendicular to the cleavage plane, the cutting operation always introduced randomly oriented crystal fragments which partly covered the surface.

Reflection	Calculated Interplanar Distance	d-values Determined Experimentally	d-values listed in A.S.T.M. Diffraction Data[24] (Molybdenite from Spokes Mine, New Jersey)	d-values listed in A.S.T.M. Diffraction Data[24] (Molybdenite from Madoc, Canada)	Reflection	Calculated Relative Intensity	I/I_0 values Determined Experimentally	I/I_0 values listed in A.S.T.M. Diffraction Data[24] (Molybdenite from Spokes Mine, New Jersey)	I/I_0 values listed in A.S.T.M. Diffraction Data[24] (Molybdenite from Madoc, Canada)
h, k, l	d				h, k, l	I/I_0			
0 0 2	6.15	6.16	6.15		0 0 2	1.39	1.40	0.8	1.0
0 0 4	3.07	3.08		3.08	0 0 4	0.021	0.02		0.2
1 0 0	2.72	2.73	2.74	2.71	1 0 0	0.060	0.05	0.7	0.5
1 0 1	2.66	2.65	2.68	2.63	1 0 1	0.384	0.19	0.2	0.2
1 0 2	2.46	2.46	2.48	2.43	1 0 2	0.086	0.07	0.9	0.5
1 0 3	2.27	2.27	2.29	2.24	1 0 3	1.000	1.00	1.0	0.8
1 0 4$\frac{1}{2}$	2.05$\frac{1}{2}$	2.04	2.04	2.04	1 0 4$\frac{1}{2}$	0.097	0.05	0.7	0.3
1 0 5	1.845	1.839	2.020	1.827	1 0 5	0.493	0.65	1.0	0.2
				1.748					0.2
				1.708					0.2
1 0 6	1.661	1.661	1.835	1.635	1 0 6	0.092	0.08	0.3	0.2
1 1 0	1.572	1.583	1.578	1.575	1 1 0	0.055	0.05	0.7	0.3
0 0 6$\frac{1}{2}$	1.538$\frac{1}{2}$	1.538	1.538	1.529	0 0 6$\frac{1}{2}$	0.345	0.30	0.9	0.7
1 1 3	1.536				1 1 3				0.3
1 0 7	1.479	1.480	1.475	1.476	1 0 7	0.024	0.02	0.2	0.2
				1.453					0.2
1 1 6	1.408				1 1 6	0.054			0.2
2 0 0$\frac{1}{2}$	1.364$\frac{1}{2}$	1.365$\frac{1}{2}$	1.369	1.368	2 0 0$\frac{1}{2}$	0.038$\frac{1}{2}$	0.02$\frac{1}{2}$	0.3	0.2
2 0 1	1.351	1.3607			2 0 1	0.0617	0.02		
2 0 2$\frac{1}{2}$	1.343$\frac{1}{2}$				2 0 2$\frac{1}{2}$				
2 0 3	1.323	1.328	1.335	1.333	2 0 3	0.302	0.15	0.7	0.3
1 1 4$\frac{1}{2}$	1.295	1.298	1.295	1.290	1 1 4$\frac{1}{2}$	0.102	0.04	0.7	0.7
2 0 4$\frac{1}{2}$	1.250$\frac{1}{2}$	1.250	1.251	1.247	2 0 4$\frac{1}{2}$	0.104	0.10	0.7	0.5
0 0 8$\frac{1}{2}$	1.230$\frac{1}{2}$	1.230$\frac{1}{2}$	1.222	1.221	0 0 8$\frac{1}{2}$	0.013$\frac{1}{2}$	0.02$\frac{1}{2}$	0.2	0.2
	1.222	1.2227			1 0 10	0.0267	0.02		

Material: MoS_2 Reaction: Cu K_α (Ni filter)

Table II—Values of d and $1/1_0$ for various crystallographic planes.

Similar results of graphite have been observed by Campbell and Koza'[16], in the study of wear of carbon brushes in dry atmospheres. They cut test specimens from a highly oriented natural graphite sample. When the cleavage surface of the specimen was sliding against the copper surface in dry nitrogen, after a small amount of initial wear, no further measurable wear was observed over a period of 9 hours. When the cleavage surface of the graphite specimen was arranged to be perpendicular to the copper surface, wear took place immediately and rapidly.

ORIENTATION OF THE MOLYBDENUM DISULFIDE SURFACE FILM: The foregoing experimental results indicate that the friction characteristics of molybdenum disulfide depend largely on crystallographic orientation. The change of the orientation of the molybdenum disulfide surface film was then studied with a Norelco X-ray diffraction spec-

trometer. The X-ray used was the Cu Kα (Ni filter) radiation.

1. DIFFRACTION PATTERN OF A COMPLETELY ORIENTED POWDER SAMPLE: In order to study the change in orientation, a diffraction pattern of a completely randomly oriented powder sample has to be obtained first to serve as the basis of comparison. Because molybdenum disulfide laminae have a tendency to lie flat on the surface, which will be seen in the following experimental results, it becomes difficult to prepare a completely randomly oriented powder sample. For this reason, reflections from the main cleavage plane (001) are usually much stronger than would be expected from a completely randomly oriented powder sample (Cf. the X-ray diffraction pattern of molybdenum disulfide obtained by Godfrey and Nelson[19]).

The method finally used by the author in preparing a randomly oriented molybdenum disulfide powder sample is as follows: One surface of a glass slide is first wetted with 50% collodion solution in amyl acetate. The molybdenum disulfide powder of a purity higher than 99 per cent is then sprinkled on the wet glass slide. The powder adheres loosely. The powder sample is then carefully put on the stage of the X-ray spectrometer without letting the powder fall off. The X-ray diffraction pattern from a sample thus prepared is shown in Fig. 4a.

From the size of the hexagonal unit cell as determined by Dickinson and Pauling[24], the interplanar distances d for various crystallographic planes can be calculated. From the known crystal structure of molybdenum disulfide, the relative intensities I/I_o of the reflected radiation from various crystallographic planes can be calculated for the perfectly randomly oriented sample. In calculating the relative intensities, the intensity of the reflection (103), instead of the strongest one, is taken as unity. This is because the strongest reflection is the second order reflection from the basal plane, and will become extremely strong when the molybdenum disulfide powder takes an oriented position.

The calculated values of d and I/I_o are listed in Table II. The d-values are arranged in the order of decreasing d. The experimentally determined interplanar distances d and relative intensities I/I_o are listed in Table II next to the calculated values. Comparing the experimentally determined results with the calculated values, one can see that they are in good agreement. The (114) reflection does not show up in Fig. 4a probably because of its very weak intensity.

There are two sets of d-values and their corresponding I/I_o-values listed in the ASTM X-ray diffraction data[27] for molybdenites from two different sources, i.e., from the Ogden Mine, New Jersey and from Haddam, Connecticut. They are also listed in Table II for close comparison. Comparing the d-values for molybdenite from Haddam, Connecticut, with the calculated d-values, one can see that the agreement is not bad except the extra reflection of the d-values, 1.749, 1.699 and 1.431. These extra lines are respectively the reflections from the planes (110), (008) and (112), and (203), caused by the Cu Kβ radiation because instead of a Ni filter, an Al filter was used with Cu radiation in determining these values. The d-values for molybdenite from the Ogden Mine, New Jersey, agree well with the calculated d-values except for a few values at low Bragg angles. The probable reason is the solarization effect caused by the extremely strong reflection from the (002) plane. The I/I_o-values for molybdenite from these two sources are visual estimates.

2. X-RAY DIFFRACTION RESULT OF SURFACE FILM: In interpreting the X-ray diffraction results, the following brief explanation might be helpful: A certain crystallographic plane of interplanar distance d can reflect X-ray of wave length λ only when the incident angle Θ of the X-ray beam satisfy the Bragg's Law, $n\lambda = 2d \sin \Theta$, where n is the order of the reflection. If a crystallographic plane lies parallel to the plane of the stage in an X-ray diffraction spectrometer, this crystallographic plane will reflect the incident X-ray when the plane of the stage makes a proper angle Θ with the incident beam, and the reflected beam is detected by an ionization chamber. The ionization chamber is so coupled with the specimen stage that only beams reflected from crystallographic planes parallel to the plane of the stage enter the ionization chamber. In a completely randomly oriented powder sample, all crystallographic plane may be assumed to have equal chance to lie parallel to the surface of the glass slide. In this case, the spectrometer register reflections from all crystallographic planes (There are limitations, such as the mechanical construction of the spectrometer. In the Norelco X-ray diffraction spectrometer, it is limited to the reflections having angles 2Θ smaller than 90 degrees.) as illustrated by Fig. 4a. If all flakes in the randomly oriented sample are rearranged to lie with a certain particular crystallographic plane parallel to the plane of the stage, only the reflections from that particular crystallographic plane will appear on the spectra and at the same time their intensity will become very strong.

Fig. 4b shows the X-ray diffraction result of the molybdenum disulfide surface film which was rubbed on the ground glass plate of 30 microinches rms surface finish with a pad of cotton. In Fig. 4b, one can see that the relative intensities of the reflections from the main cleavage plane, i.e., (002), (004), etc., are much stronger than the normal magnitudes (compare with Fig. 4a). Besides the very strong reflections from the main cleavage plane, only the reflections, (103) and (105), are barely visible. It is to be mentioned here that a spectra showing only reflections from one particular crystallographic plane, does not necessarily mean that every flake is lying with that particular crystallographic plane parallel to the plane of the stage. Therefore, the result Fig. 4b indicates that, within the volume of the sample irradiated by the X-ray, there are more molybdenum disulfide laminae oriented with their main cleavage plane parallel or almost parallel to the surface of the base than would be in the case of completely randomly oriented powder sample.

3. APPARATUS USED FOR RUBBING THE SURFACE FILM: The apparatus for rubbing a rider over molybdenum disulfide surface film was quite simple. It essentially consisted of a spherical glass rider of 6 mm diameter sliding on a rotating flat ground glass plate. The load was applied by putting weights along the vertical axis of the rider holder. A flat circular wear spot of approximately 1.5 mm diameter was made on the spherical rider by running the rider on the rotating ground glass plate. The rider could be moved radially. This movement was necessary for preparing a track which is wide enough for X-ray diffraction study.

4. EFFECT OF THE RUBBING PRESSURE: Fig. 4c shows the X-ray diffraction result of the molybdenum disulfide surface film on the ground glass plate of 30 microinches rms after the surface film has been rubbed over about ten times by the glass rider under a load of 3,000 gm. All of the originally weak reflections have disappeared into the background, and only the reflections from the main cleavage plane are left in the pattern. The action of rubbing with the glass rider under a load of 3,000 gm causes a large percentage of the molybdenum disulfide laminae in the surface film to lie with the main cleavage plane parallel or almost parallel to the surface. Fig. 4b and 4c are directly comparable in showing the effect of the rubbing pressure. The rubbing pressure exerted by the glass rider under a load of 3,000 gm is much greater than that produced by the pad of cotton. The results seem to indicate that the higher the

(Continued on page 306)

METAL CUTTING DEVELOPMENTS*

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Mr. Pigott received his degree in Mechanical Engineering from Columbia University in 1906. His experience prior to joining the Gulf Research & Development Co. in 1929 as Director of the Engineering Div., included the design, construction and operation of central steam power stations; designing and constructing industrial plants; and the manufacture of metal products. He holds more than 30 patents on devices used in hydraulic engineering, heating, oil production, and instruments, and is the author of some 40 technical papers on power plant design, fluid flow, instruments, lubrication, and superchargers.

For more than fifty years it has generally been the practice to apply cutting fluids as shown in Fig. 1, top stream. More recently, many have followed the so-called "flooding" method (both streams, Fig. 1). Some improvement was obtained in some cases, none in others.

A little thought will show what is wrong: The upper stream flows over the chip but is blocked from reaching the cutting edge by the chip and by pressures of over 50,000 p.s.i. subsisting between the cutting edge and the cut face of the work, and likewise between the chip and the top of the tool. For low cutting speeds such as used in threading, gear cutting, or for very tough or hard materials, the rate of movement is low enough so that capillary forces may act, and small quantities of cutting fluid may reach the tool edge. But for normal turning, boring and milling speeds 100 f.p.s. and higher, the cut surface is bone dry, free of oil, and the only possible function of the cutting fluid is indirect cooling. For this reason, soluble oils (which are mostly water) have done quite well.

The writer developed a new method of applying cutting oil that has produced rather surprising results. A jet, small enough in width to enter the space between the heel of the tool and the moving work, is applied from below the cutting edge as shown in Fig. 2. This jet does not splash on the bottom of the tool; in order to penetrate to the top of the wedge between tool and downward moving work, high speed is necessary.

Preliminary calculation showed that with work moving past the tool edge at 1.4 f.p.s. and higher, a very considerable friction resistance to flow subsists in the upper portion of the wedge. Consequently a jet speed is required that will, on turning at right angle under the tool contact, yield *more* static pressure than this friction loss. The optimum pressure appeared to be centering around 400 p.s.i. (250 f.p.s. jet speed) for mineral cutting oils. Fig. 3 shows the results of tests on SAE 3150 steel at 110 s.f.m., 0.011 in. feed and 0.150 depth of cut. Not much improvement is shown at the lower pressures, a sharp increase between 250 and 400 p.s.i., and thereafter a decrease again. We have since found, as might be expected, that as cutting speed goes up the jet pressure should also be increased moderately, and change of viscosity of the cutting liquid also influences the optimum pressure.

In order to avoid cross flow, the high-speed jet is made considerably smaller than the gap between tool and work. In our test cases, the tool is generally ½ in by ½ in., the space between the heel of the tool and work about 0.087 in. Calculation showed that much less fluid than the usual flow would be required for adequate cooling. All the fluid is spread on the flank of the tool in a very thin stream moving very fast. We used a jet 0.010 in. diameter at first, but have tried larger sizes and for deep cuts, multiple jets and slit orifices. At present we are generally using one or more 0.014 in. diameter jets spaced 1/16 in. to 9/64 in. or so, as necessary. A suitable test procedure was developed to measure pressures under the tool edge, shown in Fig. 4.

As would be expected, since the oil flowing up to the tool edge must turn at right angles to get out, the static pressures are obtained by the change of direction in flow. The function of this static pressure is highly important and will be discussed below.

Since the tool edge is at temperatures of 800 to 1200 F., boiling of the cutting field occurs thereon and cooling is greatly improved. Lubrication of the tool is also accomplished at any speed as proved by several indications, as follows: (1) Tool life at any normal speed is increased 5 times or more; (2) Tool life at higher than normal speeds is increased much further; for example, cutting 1020 steel at 400 f.p.m. (3.5 times normal) the tool life is about the same as at 120 f.p.m., but is 30 to 60 times the life using the old method of application; (3) Build-up is eliminated; (4) Forces for cutting and feed are reduced; (5) Finish improves from say, 200 microinches to 65; (6) The diameter of the work does not change more than .001 to .003 in. until tool breakdown; (7) The nose of the tool remains in good condition, failure taking place on the cutting edge when tested to destruction. See Fig. 5.

How does a pressure of a few hundred pounds per square inch force fluid past contacts of 50,000 to 100,000 p.s.i.? We examined the roughness of the cut face (the shoulder) versus that of the tool. The cut face roughness was about 65 microinches, that of the tool about 6-8 microinches; this means that the work is splitting to some extent ahead of the tool edge and passages of 10 to 70 microinches depth subsist, providing the necessary passages over the edge. Probably the pressures available under the edge, from the jet impact, are insufficient to drive any appreciable amount of liquid oil through these tiny conduits; but we know from the considerably increased smoke and vapor from under the tool edge that the liquid is boiling at the edge. Oil vapor can be driven through these passages by the pressures available. As soon as the vapor reaches a little distance back of the edge, the lower temperature of chip and tool recondenses this vapor and it lubricates the chip and crater as liquid oil. A further proof of lubrication is given by the dynamometer forces; the turning or tangential force is largely that of cutting and bending the chip, but there is also edge friction. When using the high speed jet this force usually drops 5 to 10% from the overhead method, for the same cut. The feed or longitudinal force is composed largely of chip friction on top of the tool; this is reduced 35

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to 50%. Improvement of finish is due to preservation and lubrication of the nose; it is very noticeable that there are no "feathers" on the work surface. The work can be safely felt with the fingers at 400 s.f.m.

Cooling is immensely improved. With the old system flows of 0.75 to 1.50 gal. per cubic inch metal removed produce 30° to 35° cooling of the tool edge, below the dry cutting temperature. With the high-speed jet 0.03 to 0.07 gal. per cubic inch produces cooling from 75 to 160 F. below dry cutting, depending on flow and cutting speed.

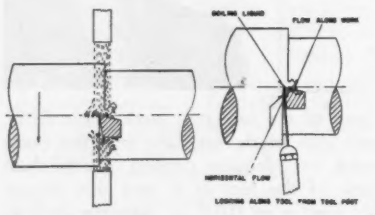

Fig. 1 (left) Flooding.

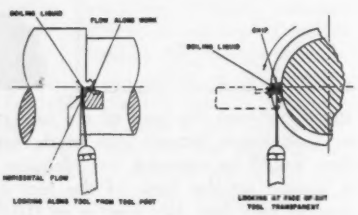

Fig. 2 (center & right) Jet location and flow path.

Apparently the direct cooling in thin films at high speed improves edge cooling 2 to 3 times with about 1/20 the oil usually circulated by overhead flooding, or 40 to 60 times the cooling per gallon circulated. There is usually no color in the chips, which can generally be caught in the hand, without discomfort, as they come off the tool, when using high-speed tools and normal speeds.

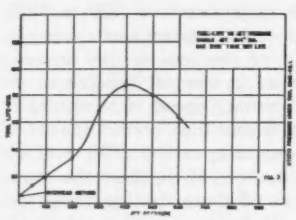

Fig. 3 (left) Tool-life vs. jet pressure.

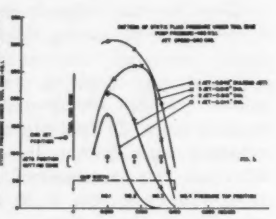

Fig. 4 (right) Pattern of static fluid pressure under tool edge.

Looking at test results, Table I gives some of our laboratory results. In these cases the Hi-Jet performance is compared to the best conventional cutting oil or emulsion. The successful cutting of titanium appears to be solved. To everybody this material has hitherto been a bad headache.

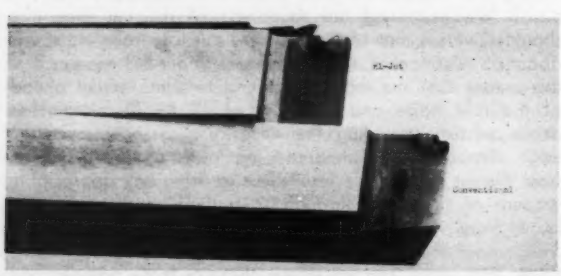

Fig. 5 Character of tool failure—top view.

The results of other tests are shown in Fig. 6 which compares the Hi-Jet and overhead performance of five cutting fluids. Dry cutting results are also included for comparison. Oil "E" is a conventional cutting oil which, when applied with Hi-Jet, gives results worse than those obtained with the overhead method. Oil "D" is another conventional cutting oil which, while giving about the same results as oil "E" with the overhead method, produces improved tool life when applied with Hi-Jet. Oil "C" is a conventional cutting oil which is the best of the conventional oils tested with both Hi-Jet and overhead. Oil "B" is an experimental oil formulated for use with Hi-Jet which is very satisfactory, but does not give good results with the overhead method. Oil "A" is our present Hi-Jet oil which, as may be seen on the figure, gives best results with Hi-Jet and equals the performance of the commercial cutting fluids when used in the overhead method. Over 60 conventional and experimental oils have been tested, including some competitive oils and the above results are typical of our findings with the conventional oils.

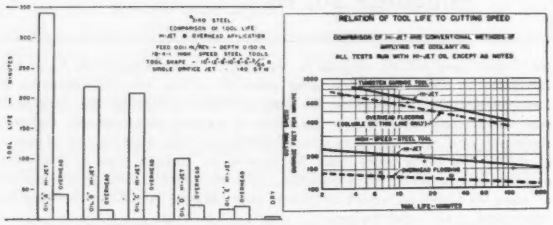

Fig. 6 (left) #3140 Steel—Comparison of tool life—Hi-Jet and overhead application.

Fig. 7 (right) AISI 4147 (INT) Steel—Test log lathe turning operation.

The nature of the cutting fluid has an influence that must not be disregarded. We quickly found that many of the highly favored- and highly compounded-oils became chemically active at the tool-edge temperature and are unsatisfactory. The oil developed for this system works best therein and in addition, is suitable for use in the machine tool bearings or in a hydraulic system connected with it. Present commercial emulsions in the ordinary mixes (5% oil, 95% water) are not very satisfactory in this concentration; they give more tool life than the overhead system with

TABLE I — TEST RESULTS AT GULF RESEARCH & DEVELOPMENT CO. OVERHEAD FLOOD WITH CONVENTIONAL COOLANTS VS. HI-JET WITH GULF HI-JET OIL

Steel	Cutting Conditions	Material	Tool Shape	Oil	Overhead Flood Test Life Time, Min.	Oil	Hi-Jet Test Life Time, Min.	Any Time, Min.
SAE 3130	110 sfm 0.011 in.feed 0.150 in. depth of cut	18-4-1 High-speed steel	10,12,0 10,6,6,3/64	Mineral Lard Sulphurized Mineral	10.00 12.60	Gulf Hi-Jet 011	377	4.3
AISI 3140	140 sfm 0.011 in. feed 0.150 in.depth of cut	18-4-1 High-speed steel	10,12,0 10,6,6,3/64	Sulphurized Mineral Sulphurized Lard Sulphurized Mineral Sulphurized Lard Mineral Lard	9.3 26.0 16.0 23.0	Gulf Hi-Jet 011	339.9	8.9
SAE 1020	212 sfm 0.013 in. feed 0.125 in. depth of cut	18-4-1 High-speed steel	0,6,6,6 8,8,1/32	Soluble	1.37	Gulf Hi-Jet 011	39.9	9.5
AISI 416 Stainless	360 sfm 0.011 in. feed 0.150 in. depth of cut	18-4-1 High-speed steel	10,12,0 10,6,6,3/64	Mineral	9.4	Gulf Hi-Jet 011	163.4	
SAE 1020	400 sfm 0.011 in. feed 0.088 in. depth of cut	18-4-1 High-speed steel	10,12,0 10,6,6,3/64	Mineral	3.33	Gulf Hi-Jet 011	80.9	2.0 min.at 395 sfm
SAE 1020	375 sfm 0.011 in. feed 0.088 in. depth of cut	18-4-1 High-speed steel	10,12,0 10,6,6,3/64	Mineral	4.0	Gulf Hi-Jet 011	125	2.0 min.at 395 sfm
AISI 3245	300 sfm 0.011 in. feed 0.088 in. depth of cut	18-4-1 High-speed steel	10,12,0 10,6,6,3/64	Soluble	1.9	Gulf Hi-Jet 011	90	2.0 min.at 370 sfm 0.5 min.at 800 sfm
Titanium Flawed Bar	100 sfm 0.011 in. feed 0.088 in.depth of cut	Carmet CA-2	0,5,7,7 6,6,1/64			Gulf Hi-Jet 011	47.4	3.44 min.at 100 sfm
Titanium	180 sfm 0.011 in.feed 0.088 in.depth of cut	Carmet CA-2	0,5,7,7 6,6,1/64			Gulf Hi-Jet 011	4.4	3.44 min.at 100 sfm

TABLE II — TESTS ON INCONEL TPM

Material	Cutting Conditions	Tool	Oil	Overhead Flood Pieces	Oil	Hi-Jet Pieces
Inconel TPM	66.5 sfm 0.0065 in. feed 0.220 in.depth of cut	Tungsten Carbide	Soluble	57	Hi-Jet	275
Inconel TPM	66.5 sfm 0.0065 in. feed 0.220 in. depth of cut	Cast Alloy Co 48 Cr 30 W 18 Ta 5	Hi-Jet	Time, Min. 7.9	Hi-Jet	Time, Min. 64
Inconel TPM	52 sfm 0.0065 in. feed 0.220 in. depth of cut	Cast Alloy Co 48 Cr 30 W 18 Ta 5	Hi-Jet	77.0	Hi-Jet	280
Inconel TPM	52 sfm 0.0065 in. feed 0.220 in. depth of cut	High-Speed Steel Mo 8 Cr 4 V 1 W 1-1/2	Hi-Jet	2.5	Hi-Jet	47
Inconel TPM	41 sfm 0.0065 in. feed 0.220 in. depth of cut	High-Speed Steel Mo 8 Cr 4 V 1 W 1-1/2	Hi-Jet	39	Hi-Jet	230

any fluid, but only 35 to 40% of the life attained with Hi-Jet oil. However, we have found that raising the concentration to perhaps 20% oil gives very satisfactory results

(Continued on page 308)

Symposium on Bearing Fatigue, Part I

MATERIAL SPECIFICATIONS FOR OIL-FILM BEARINGS*

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ABSTRACT: This paper describes the selection of metal compositions for various application in oil-film bearings, enumerating the factors and functional differences that influence the choice of available bearing materials in their use, some of which are in the final period of development.

Before a satisfactory analysis of fatigue in the metal of a plain oil-film or boundary-film journal bearing may be made, the bearing material must first be capable of operating or running successfully under the load conditions imposed. The main requirements,[1,2,3] including fatigue, that must be met are: 1. Score Resistance (Anti-Seizure); 2. Deformability: a. Comformability; b. Embedability; 3. Fatigue Strength; 4. Corrosion Resistance.

As will be shown, the listed bearing requirements are interrelated in some respects. However, bearings are subjected to such varied influences that the selection of a metal for a particular application is usually a compromise, and the best choice is made by recognizing the maximum requirement and then selecting a commercial alloy having the desired qualities.

SCORE RESISTANCE (Anti-Seizure): Long before Isaac Babbitt obtained a patent (#1252, July 19, 1839) on a journal box containing an alloy of tin, antimony, copper and lead, for use as a plain bearing liner (Fig. 1), and up to the present time, this type of bearing metal, now widely known as babbitt, has been predominantly used with very satisfactory results. Its chief weakness in cyclic loaded bearings is low fatigue life. One of its attributes is a very excellent anti-seizure characteristic, which is now being utilized as a very thin overlay on the most modern types of high-load-capacity plain bearings. These are also covered in a later section. Isaac Babbitt even foresaw the effectiveness of thin layers as in his description of the journal box he said, ". . . the ledges will effectively check the tendency in the metal to spread from the weight and friction of the load. When the ledges are not used the coating of the composition metal should be but thin."

Sufficient tests[4,6] and experiences[3,5] have now been accumulated to provide some order to the many available bearing metals, as shown in Table I. Improved resistance to scoring (not to be confused with cutting or grooving of the metal by dirt and other foreign material) usually has been found to be obtained at the expense of hardness and strength. Increase in lead content and a decrease in tin and zinc improves the resistance to seizure of the brasses and bronzes. A general classification as listed in this table does not allow the inclusion of special bearing materials, such as the tri-metal types of plain bearings. The aluminum alloy, and also silver and copper, may be indented as in Fig. 2 with a fine grid pattern[3] and filled with lead alloy, or simply overplated with tin or lead alloy, to improve their score resistance and to be nearly equivalent to the white metals in this respect.

DEFORMABILITY: When loads are applied to a structure, the bearings or journals deflect, often producing edge contact. If a conformable bearing metal is selected for the design, local deformation takes place and the surface is automatically adjusted without producing high temperatures which result in a burned-out or fatigued bearing.

Futhermore, when dirt enters the clearance space, it must embed in the bearing metal or cut a groove around the shaft. Solid particles (particularly silica and other hard metallic oxides) are a prime cause of journal wear and bearing overheating. When copper-lead, bronze, or aluminum alloy bearings are used, it may be good practice to harden the journals to reduce the journal wear rate. White metals are preferred where dirt conditions are not controllable, even though such a compromise may involve a lowering of the fatigue life.

Frequently, the ability to embed when dirt is present and to conform when distortion is present are treated as separate characteristics. In practice, a material responds about equally to each of these influences. Table II shows this characteristic as deformability.

Table III summarizes the principal results of relative embedability tests on a number of different commercial bearing materials with an abrasive-containing lubricant.[7] The most striking fact is the beneficial effect of the lead-base babbitt overlays. This is seen by comparing identical materials with and without overlay, such as items 3 and 6, 10 and 11, 12 and 13. It is shown that a relative embedability or dirt-tolerance improvement of from 10 to 100 times may be had, depending upon the thickness of the overlay and the character of the basic bearing material underlying the steel backing.

FATIGUE STRENGTH: Bearing fatigue, the subject of the symposium, shows up in bearing metals the same as in steel; cracks will develop if the load and temperature conditions are sufficiently severe. Well known examples are connecting rod and main bearings in internal-combustion engines, which are subjected to rotating loads, and piston pin bushings, which are subject to alternating loads. When a large enough portion of bearing surface has fatigued, the oil film conditions deteriorate to the point where failure by overheating quickly follows.

The operating conditions, especially time and temperature, the design of the bearing supports, and the amount of journal flexure have so much influence that it is difficult to state specific loads. However, in Table II a guide to the maximum load is given.

The methods and techniques of manufacturing these metals for plain bearings affects their fatigue life. For example, performance of the lead-base and tin-base babbitts has been improved by precision manufacturing to obtain thin layers.

* Sponsored by the Technical Committee on Bearings and Bearing Lubrication and presented at the 7th Annual Meeting, ASLE, Cleveland, April 8, 1952.

The fatigue strength of aluminum alloys reach their highest value when securely bonded to a steel back[14]. However, long life is found with solid aluminum alloy bearings[8] when the shells can be made quite thick and the temperature rise is not sufficient to cause permanent set.

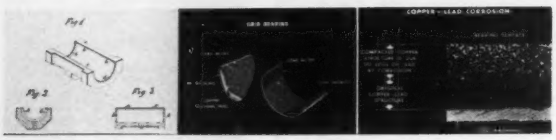

Fig. 1 (left), Fig. 2 (center), Fig. 3 (right).

Silver is a specialized bearing material used in aircraft engines because of its high fatigue strength[3].

The aluminum alloy, silver, and copper-lead bearings are, in the latest practice, used with a thin overlay (.001" thickness or less) or lead-base babbitt to aid in the "breaking in" process and for more tolerance to dirt and journal imperfections.

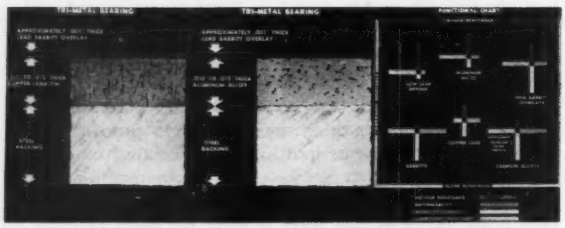

Fig. 4 (left), Fig. 5 (center), Fig. 6 (right).

CORROSION RESISTANCE: Bearing metals of copper, lead, copper-lead and cadmium that are unalloyed are subject to attack by lubricating oils containing acids, or which become acidic through oxidation. (See Fig. 3). Silver is most seriously attacked by compounds containing active sulphur in lubricating oils. Alloying zinc with copper greatly retards corrosion of copper by sulphur, and can be used in applications calling for E. P. gear oils. However, as pointed out in Table I, addition of zinc to copper impairs its anti-friction or anti-seizure properties.

TABLE I

TABLE II

TABLE III

Lead-base alloys containing over 3% tin or not less than 1% tin and suitable amounts of antimony, arsenic or indium are protected from acidic lubricants. Cadmium bearings can be protected by very small amounts of indium[9, 10].

In Table IV, the common bearing metals are classified as to their resistance to corrosion. The table should be used with discretion since the "intermediate" and "corrodible" alloys may reverse their activity in the presence of certain lubricants. The almost universal practice of marketing lubricating oils with detergents and inhibitors has greatly diminished the possibility of bearing corrosion. However, lubricant specialists should be consulted as to the type of oil to be used.

COST: The selection of the proper metal for a bearing may, more often than not, be influenced by cost. There are, no doubt, instances where the cost of a single bearing may be small as compared to the total cost of the machine. This is not an engineering justification for deliberately using silver if a simple bronze bushing would be satisfactory. The approximate order of increasing cost is shown in Table V.

TABLE IV

CORROSION RESISTANCE OF BEARING METALS

NON-CORRODIBLE
- Aluminum alloys
- Tin base babbitt
- Lead base babbitt (Properly alloyed)
- Cadmium-Indium alloys
- Bronze (Low lead)

INTERMEDIATE
- Bronze (High lead)
- Copper-lead
- Alkali-hardened lead
- Silver (Attacked by sulphur under certain conditions)

CORRODIBLE
- Cadmium alloys

TABLE V

APPROXIMATE ORDER OF INCREASING COST

- Bronze Bushings
- Lead Base Babbitt
- Tin Base Babbitt
- Sintered Copper—Nickel Filled with Lead Babbitt
- Aluminum Alloys (Solid)
- Cadmium Alloys
- Copper-Lead
- Copper-Lead with Thin Overlay
- Aluminum Alloy with Thin Overlay
- Silver with Thin Overlay

COMMERCIAL BEARING METALS — BABBITT METALS: Generally, in choosing a bearing metal, the babbitts should be given the first consideration. Tin-base and lead-base babbitts are easily handled and, with the exception of simple bronze bushings, have the lowest costs. It is necessary with either the tin-base or lead-base babbitts to have a minimum lining thickness properly bonded to a steel back, consistent with the embedability and conformability requirements of the application, to secure maximum load-carrying ability and fatigue life at elevated temperatures.

TABLE VI

S.A.E. STANDARD BABBITT METALS

	SAE 10	SAE 11	SAE 12	SAE 13	SAE 14	SAE 15
Tin	90.0	86.0	89.25	6.5.5.5	9.25-10.75	0.9-1.25
Antimony	4.0-5.0	6.0-7.5	7.0-8.5	9.25-10.75	14.0-16.0	14.50-15.50
Lead	0.35	0.35	0.35	80.0	75.0	Remainder
Copper	4.0-5.0	3.0-4.5	3.25-3.75	6.00	2.50	0.60
Iron	0.08	0.08	0.08			
Arsenic	0.10	0.10	0.10	0.00	0.60	0.80-1.10
Bismuth	0.00	0.00	0.00			
Zinc and Aluminum	None	None	None	None	None	None

TABLE VII

S.A.E. STANDARD COPPER-LEAD BEARING ALLOYS

	SAE 48	SAE 49M
Copper	67.0-71.0	66.0-70.0
Lead	30.0-33.0	30.0-34.0
Silver	1.5	
Tin	0.1	
Phosphorus	0.005	
Iron	0.35	0.35
Tin		0.00
Other elements	0.15	0.3

TABLE VIII

S.A.E. STANDARD BRASS AND BRONZE BEARING ALLOYS

	SAE 40 Cast Lining	SAE 62 Cast Lining	SAE 64 Cast Lining	SAE 68A Cast Lining	SAE 791 Rolled or Wrought	SAE 792 Cast Lining	SAE 793 Cast Lining	SAE 794 Cast Lining	SAE 795 Rolled or Wrought
	COMPOSITION IN PERCENTAGE								
Copper	84.0-86.0	86.0-89.0	78.0-82.0	86.0-89.0	remainder	remainder	83.0 min.	68.5-75.5	88.0-92.0
Tin	4.0-6.0	9.0-11.0	9.0-11.0	0.50	3.5-4.5	9.0-11.0	3.5-4.5	3.0-4.0	0.25-0.75
Lead	4.0-6.0	0.30	8.0-11.0		3.5-4.5	9.0-11.0	7.0-9.0	21.0-25.0	
Zinc	4.0-6.0	1.0-3.0	0.75		1.5-4.0	0.5	4.0	3.0	remainder
Antimony			0.50			0.5			
Nickel	1.0	1.0	0.50			0.5			
Iron	0.30	0.15	0.15	2.5-4.0	0.1	0.35	0.35	0.35	0.1
Aluminum	0.005	0.005	0.005	0.5-2.5					
Phosphorus			0.25						
Silicon	0.005		0.005						
Other elements					0.2	0.4	0.3	0.4	0.2

TABLE IX

ALUMINUM BEARING ALLOYS

	COMPOSITION IN PERCENTAGE		
	XB-805	750	300750
Aluminum	Balance	Balance	Balance
Tin	6.5	6.5	0
Nickel	1	0.5	0
Copper	1	1	0
Silicon	0	2.5	4
Cadmium	0	0	1.0

TABLE X

S.A.E. STANDARD CADMIUM BEARING ALLOYS

	SAE 10	SAE 10M
	COMPOSITION IN PERCENTAGE	
Cadmium, minimum	98.4	98.35
Nickel	1.0-1.6	0
Silver	0.01	0.5-1.0
Copper	0.30	0.4-0.75
Tin	0.02	0.01
Lead	0.05	0.02
Zinc	0.00-0.15	0.02

Babbitt linings of .003" to .008" thickness are common practice in commercial automotive engines with .002" the low limit on babbitted-in-place bearings. Large stationary engines use thicknesses from .020" to .060". The SAE 10 (tin base) and SAE 14 (lead base) are typical. There are numerous variations in composition, but it has been found difficult to show differences in fatigue life if the lining is in the order of .002" to .004" thick. Table VI shows the SAE Standard Babbitt Metals.

COPPER-LEAD ALLOYS: Copper-lead bearings are backed by steel to give strength and rigidity. They are widely used where loads are higher than can be carried on babbitts, and are made by casting[11] or by the sintering

(Continued on page 312)

Symposium on Bearing Fatigue, Part II

TIRED BEARINGS — REASONS AND REMEDIES*

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Mr. Palsulich graduated from the Guggenheim School of Aeronautics College of Engineering, New York University, in 1933 with the degree of B.S. in Mechanical Engineering, receiving his M.E. degree in 1938. During 15 years with Wright Aeronautical Corp. he engaged in aircraft engine design and test, specializing on bearings, seals, and engine lubrication problems. In 1949 he joined The Cleveland Graphite Bronze Co. as Staff Engineer handling aircraft bearing problems; he has since been promoted to Assistant Chief Engineer—Aircraft and Allied Products. He is a member of SAE, Tau Beta Pi, and the NACA Subcommittee on Lubrication and Wear.

ABSTRACT: This paper describes the composition and construction of sleeve bearings in common use. The effects of design and operating conditions upon fatigue are illustrated. The comparative resistance to fatigue of various bearing materials are shown in tabular form. Some not too well recognized types of fatigue are discussed. Erosion as a specialized form of bearing fatigue is described and remedies proposed for its elimination. Fatigue as an aftermath of lead corrosion in copper-lead bearings is illustrated.

COMPOSITIONS: Fatigue in sleeve bearings does occur in many instances, unfortunately, but it can usually be avoided by proper attention to design. Sufficient data have been accumulated during the past decade on bearing materials and construction and engine design to enable the bearing design specialist to avoid the pitfalls and the resultant penalty of "tired" bearings.

Due to the endless variety of design and operating conditions to be met in sleeve bearing applications, no handbook has yet been evolved which will enable the newcomer to learn the art in five easy lessons. As a small step in this direction, this paper will describe several features of bearing construction and design which should aid in avoiding bearing fatigue.

Numerous compositions of bearing materials have been "cooked up" in the quest for inherently higher fatigue life than possessed by babbitts. Some of the ones in common use are shown in Table I. In the upper section are listed the "bimetals." Experience soon showed that increased fatigue resistance was gained at the expense of good frictional properties, and it was found necessary to resort to the use of hardened journals. The various bronzes, copper-leads, silver and aluminum are in this category.

To regain the excellent frictional and embedability properties of the babbitts without surrendering the improved fatigue resistance of these newer materials, a thin layer of the soft babbitt or babbitt-like substance was super-imposed on them, resulting in the "tri-metals" shown in the lower section of Table I.

Present manufacturing methods permit the fabrication of trimetals either by casting or electro-plating the surface layer. The electro-plated surface layers or "overlays" commonly used are also shown. The minor constituents in these overlays serve to inhibit the lead against corrosion and to harden and increase their fatigue strengths. These overlays are generally meant to be more-or-less permanent running surfaces and not merely run-in surfaces.

* Sponsored by the Technical Committee on Bearings and Bearing Lubrication and presented at the 7th Annual Meeting, ASLE, Cleveland, April 8, 1952.

FATIGUE RESISTANCE: The relative fatigue resistances of the commonly used bearing materials, starting with the highest values, are as follows: 1. Trimetal (F-77), 2. Leaded bronze with .001" overlay, 3. Silver with overlay, 4. Aluminum alloy (solid construction), 5. Copper-lead (F-12), 6. Lead-base Micro Babbitt (.004" thick), 7. Tin-base Micro Babbitt (.004" thick), 8. Cadmium alloys, 9. Lead-base Babbitt (.020" thick), 10. Tin-base Babbitt (.020" thick).

The gap between the babbitts and the bronzes and silver is quite large. The fatigue resistance of both the solid aluminum and the steel-backed aluminum is somewhat better than that of the copper-leads but lower than that of the bronzes and silver.

CONSTRUCTIONS: To permit the achievement of the highest fatigue life of the materials available, it was found necessary to use them in thin layers. The common construction of the bimetal and trimetal bearings is shown in Table II. Note the very thin layers of babbitt used. A very considerable increase in the fatigue life of babbitt bearings was obtained by reducing the lining thickness to .002/.005". The variation of fatigue life versus babbitt lining thickness is shown in Fig. 1, as established by bench rig, dynamometer and field tests.

The ultimate in fatigue life of babbitt bearings is obtained in the construction with .001" lining thickness. This can be supplied in either a cast or electro-plated layer. In this case, the bearing is precision finished so that no further machining is necessary.

The bronzes, copper-leads and silver lining thicknesses are usually in the order of .020/.025" and are not as critical as the thickness of babbitts due to inherently higher physical strengths of these materials. There is revived interest in steel-backed aluminum alloys, and considerable effort is being expended in developments involving bonding techniques and achievement of the optimum composition from the viewpoint of good frictional properties and embedability.

TYPES OF FATIGUE: Fatigue failures of sleeve bearings may be due to many causes. One of the common causes is localized edge loading. This can result either from excessive shaft deflection or an hour-glass contour due to improper deflection or an hour-glass contour due to improper machining of the shaft or housing. A typical overall fatigue failure may be due to overstress. High operating temperatures are often a contributing cause.

In Fig. 2 is shown a fatigue failure of a copper-lead bearing more commonly called "spalling." The basic cause was removal of lead from the surface layer by oil corrosion and subsequent fatigue of the remaining, fragmentary copper layer. The obvious remedy is to select lubricants and adjust engine operating conditions so that they will not be conducive to the formation of harmful acids.

A special form of fatigue common in varying degrees to many bearing materials when operated under conditions of high impact loading shown in Fig. 3.

This phenomenon frequently occurs in high speed planetary reduction gear pinion bearings as the result of shock loads caused by the gear tooth mating action. In some cases the forces have been severe enough to cause erosive fatigue to a depth of 1/32" in a steel journal. The same type of damage has been found in connecting rod bearings in Diesel engines and in master rod bearings in aircraft engines. The basic forces causing damage in these instances appear to be the high peak gas pressures associated with detonation.

The theory behind this phenomenon is not too clearly defined at present but some studies are underway which may lead to a better understanding of the matter. One of the present theories is that bubbles form in the cavitation region where the absolute pressure has dropped below the vapor pressure of the liquid. These vapor-filled bubbles then travel to some other point in the bearing where the

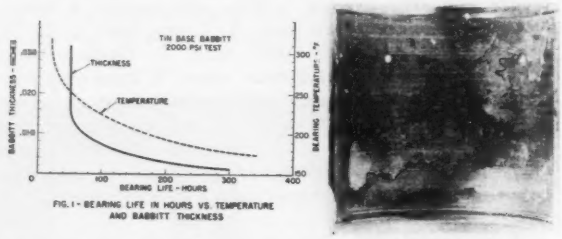
TIN BASE BABBITT
2000 PSI TEST
THICKNESS
TEMPERATURE
BABBITT THICKNESS - INCHES
BEARING TEMPERATURE - °F
BEARING LIFE - HOURS
FIG. 1 - BEARING LIFE IN HOURS VS. TEMPERATURE AND BABBITT THICKNESS

Fig. 1 (left) Bearing life in hours vs. temperature and babbitt thickness.

Fig. 2 (right) Surface fatigue resulting from lead corrosion.

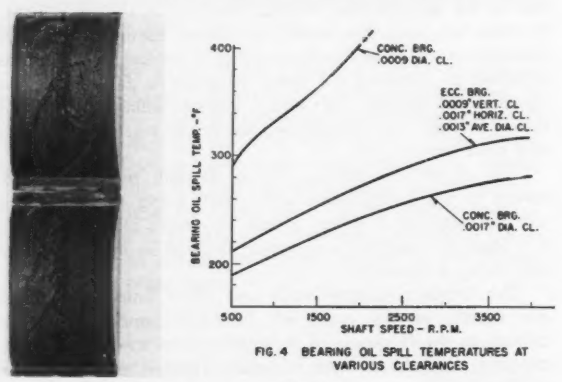
CONC. BRG. .0009 DIA. CL.
ECC. BRG. .0005" VERT. CL. .0017" HORIZ. CL. .0013" AVE. DIA. CL.
CONC. BRG. .0017" DIA. CL.
BEARING OIL SPILL TEMP. - °F
SHAFT SPEED - R.P.M.
FIG. 4 BEARING OIL SPILL TEMPERATURES AT VARIOUS CLEARANCES

Fig. 3 (left) Cavitation erosion type of fatigue.

Fig. 4 (right) Bearing oil spill temperatures at various clearances.

velocity head decreases and the pressure head increases, causing collapse of the bubbles. The rate of collapse actually determines the pressures produced, and it is commonly believed that pressures on the order of 50,000 p.s.i. can be produced over microscopically small areas. The repeated hammering, due to the collapsing bubbles, will fatigue and remove sections of the bearing materials.

The problem has been attacked in a practical manner, and some relief measures may be applied as follows: 1. At least two investigators, our own Research Division and MIT, have duplicated the phenomenon on bench test rigs (magneto-striction vibrators) and have catalogued the commonly used bearing materials in regard to their resistance to cavitation erosion. These data have not been published but are taken into account by bearing engineers when selecting materials for applications wherein cavitation erosion is likely to occur. 2. It has been found that high bearing operating temperatures adversely affect cavitation erosion. The lowering of the lubricant viscosity appears to be the significant factor rather than the high temperature per se. Under given test conditions the bearing damage was five to ten times as great at 300 F. as at room temperature, depending on the material. Use of as high as possible viscosity of the lubricant at operating temperature is, therefore, indicated. 3. In some cases, a solution to the problem was found in a rearrangement of oil hole locations, grooving, etc. 4. In view of the magnitude of the forces causing the damage, the best approach to the problem may be an attempt to eliminate the basic cause. Two methods have been used successfully in the case of sleeve bearings installed in gears: (a) DAMPENING THE SHOCK IMPULSES:

(BIMETALS)

MATERIAL DESIGNATION	SPEC	NOMINAL COMPOSITION - % COPPER	TIN	LEAD	ZINC	ANTIMONY	SILVER	OTHERS	FORM OR CONSTRUCTION
LEAD BASE BABBITT (F-11)	SAE 14	1/4	10	74		15		1/2 ARSENIC	SPLIT BUSHINGS, MADE FROM STRIP OR GRAVITY CAST BEARINGS
TIN BASE BABBITT (F-1)	SAE 12	3 1/2	89			7 1/2			SPLIT BUSHINGS AND HALF BEARINGS MADE FROM STRIP OR GRAVITY CAST BEARINGS
LEADED BRONZE (F-5)	SAE 792	80	10	10					SPLIT BUSHINGS, MADE FROM STRIP OR GRAVITY CAST BEARINGS
LEADED BRONZE (F-7)	SAE 794	72	3	23	2				SPLIT BUSHINGS AND HALF BEARINGS MADE FROM STRIP OR GRAVITY CAST BEARINGS
COPPER LEAD (F-12)	SAE 480	65		35					SPLIT BUSHINGS AND HALF BEARINGS MADE FROM STRIP OR GRAVITY CAST BEARINGS
COPPER LEAD (F-14)	AMS 4820	71		28			1		GRAVITY CAST BEARINGS
SILVER (F-21)	AMS 4815						99 3/4		ELECTRO PLATED BEARINGS USED WITH LEAD-TIN OR LEAD INDIUM OVERLAY
ALUMINUM (F-81)		1	6					1-NICKEL BAL-AL	CAST OR FORGED - SOLID CONSTRUCTION

(TRIMETALS)

MATERIAL DESIGNATION	SPEC	NOMINAL COMPOSITION - % COPPER	TIN	LEAD	OTHERS	FORM OR CONSTRUCTION
CAST TRIMETAL (F-17)		80	10	10		CENT. CAST BEARINGS WITH CAST TIN BASE BABBITT OVERLAY
PLATED TRIMETAL (F-77)	SAE 49	75	1/4	24 1/2		STRIP BUSHINGS & HALF BEARINGS MADE FROM STRIP WITH F-77 PLATED OVERLAY
CAST TRIMETAL (F-88)	AMS 4824	75	1	24		GRAVITY CAST BEARINGS WITH CAST TIN BASE BABBITT OVERLAY

(OVERLAYS)

MATERIAL DESIGNATION	SPEC	NOMINAL COMPOSITION - % COPPER	TIN	LEAD	INDIUM	FORM OR CONSTRUCTION
LEAD-TIN-COPPER (P-77)		2 1/2	10	87 1/2		ELECTRO CO-DEPOSIT
LEAD-TIN (P-78)			10	90		ELECTRO CO-DEPOSIT
COPPER-TIN (P-89)		3	97			ELECTRO CO-DEPOSIT
LEAD-INDIUM (P-82)				94	6	SEPARATE ELECTRO DEPOSITS FOLLOWED BY DIFFUSION TREATMENT

Table I. Commonly used bearing materials in automotive, diesel, and aircraft applications.

MATERIAL	LINING THICKNESS (NOMINAL)	OVERLAY (NOMINAL)	BACKING
BIMETALS			
CONVENTIONAL TIN OR LEAD BASE BABBITTS (F-1 & F-11)	.020"	NONE	SAE 1010-1020 STEEL THICKNESS VARIED TO SUIT INDIVIDUAL DESIGN
MICRO, TIN, OR LEAD BASE BABBITTS (F-1 & F-11)	.002/.005" RODS .004/.007" MAINS	"	"
LEADED BRONZES (F-5 & F-7)	.020"	"	"
COPPER LEADS (F-12 & F-14)	.020"	"	"
TRIMETALS			
CAST TRIMETAL LEADED BRONZE (F-17)	.025"	.005"	SAE 1010-1020 STEEL THICKNESS VARIED TO SUIT INDIVIDUAL DESIGN
CAST TRIMETAL LEADED BRONZE (F-88)	.013"	.001"	"
PLATED TRIMETAL COPPER-LEAD (F-77)	.013" STRIP METHOD .025" GRAVITY CAST	.001" .001"	"
PLATED TRIMETAL SILVER (F-21)	.013"	.001"	"

Table II. Construction of bimetal & trimetal bearings.

This can be accomplished to a degree by the introduction of an extra oil film between the bearing and the source of shock impulses. Either a full-floating bearing or a bearing designed to float in its housing but constrained against rotation would theoretically attain the objecive. Experience has shown, however, that full-floating bearings do not float under all conditions of operation and their theoretical advantages cannot be realized. A bearing designed to rotate with the gear but with an oil film clearance on the O.D. and restrained against rotation has been found to give improved performance under conditions which would severely damage a normal, pressed-in type of bearing. (b) SMOOTHING-UP GEAR ACTION: This can be accomplished

(Continued on page 313)

Symposium on Bearing Fatigue, Part III

ROLE OF THE LUBRICANT IN BEARING FATIGUE*

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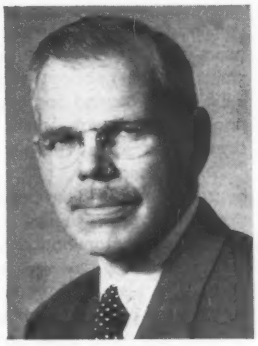
Technical Service Dept.
Brooklyn, N. Y.



Mr. Carmichael received his B.S. degree in Chemistry from Oberlin College and his M.S. degree in Chemistry from the University of Michigan. After three years on the teaching staff of the Chemistry Dept. of the University of Michigan, he joined the Socony-Vacuum Oil Co., Inc., as research chemist. Until July of this year he supervised the Lubricants Div. of the Technical Service Dept., at which time he became Technical Assistant to the Director.

Mr. Purdy received his M.E. degree from Cornell University. He was employed by the Detroit Edison Co., and as Associate Editor of *Power*, prior to joining the Socony-Vacuum Oil Co., Inc., in 1939 as Staff Engineer of the Lubricating Dept.

ABSTRACT: Bearing fatigue failures, in general, are not influenced directly by the lubricant. However, by preventing excessive bearing temperature rises, by wearing-in or effecting highly polished surfaces, by protecting against corrosion and by sealing against contaminants, the lubricant may often play an indirect rule in minimizing metallic fatigue. This paper deals with the treatment of such factors insofar as bearing fatigue failures are concerned. Examples are cited, taken both from the literature and from actual field experience, to illustrate the effectiveness of the lubricant in this respect.

Bearing fatigue failures, in general, are not influenced directly by the lubricant. For this reason, it is not uncommon for the statement to be made that this type of bearing failure is metallurgical in nature and that the condition cannot be prevented by lubricating oil or grease. Although this statement applies in by far the majority of cases, the lubricant may often play an indirect role in minimizing fatigue.

At the outset, it should be stated that the term "bearing fatigue" as used in this paper refers to a metal failure which results from its having been subjected to a large number of cyclic stresses below the tensile strength of the metal. In this discussion, the following three different causes of fatigue are recognized: (1) Localized stresses due to roughnesses, which may result in pitting: Metal surfaces are not perfectly smooth and even when finish is good, surface irregularities exist. When two such dry surfaces are in contact under load without motion, high spots deform permitting more and more surface contacts until the load is supported. Under these conditions, the load is not evenly distributed, but is concentrated on the high spots which may be stressed above their endurance limit, depending upon the load and smoothness of finish. If cyclic heavy loads are imposed on these surfaces, the high spots are repeatedly subjected to high localized stresses until fatigue of the metal causes minute particles to break away and form pits. Where full fluid films of lubricant are interposed between the surfaces, the high spots are not in contact and this type of fatigue pitting will not occur. Under conditions of boundary lubrication, the films of lubricant are so thin that some of the high spots touch, and repeated loading will cause the fatigue thereof and the formation of pits. This type of fatigue may occur at the pitch line of gears and in anti-friction bearings, but will not develop in boundary lubricated plain bearings, because sliding action wears away the high spots before fatigue occurs. As will be explained later, the lubricant can sometimes protect against pitting from this cause under conditions of boundary lubrication. (2) Localized stresses due to stress raisers, such as cracks, pits and dents, which may lead to fatigue: Bearing surfaces may contain invisible cracks as a result of improper grinding, or cracks may develop subsequently because of local heat stresses. These cracks, and pits which may be caused by corrosion, tend to produce localized stresses that are materially higher than elsewhere in the metal. When high local stresses are cyclic and above the endurance limit of the metal, the surface cracks may extend and pits develop further until failure occurs. The lubricant can be effective against this type of fatigue only insofar as it can prevent the formation of the initial stress raisers. (3) Subsurface stresses due to high compressive loads, which may result in spalling: Because of metal elasticity, when two heavily loaded surfaces roll or slide together, a wave tends to form and move ahead of the area of contact. Under heavy loads, the metal below the surface of the wave may be stressed beyond its endurance limit. When subjected to repeated stresses of this nature, subsurface fatigue failures eventually occur and relatively large flakes of metal are rolled or spalled from the surfaces. Our experience indicates that the lubricant is ineffective in protecting against this type of fatigue failure, except insofar as it is able to prevent bearing temperatures from rising excessively.

In the present discussion, scored or galled bearing surfaces, which result from failure of the oil film to carry the load, either because the operating conditions were abnormally severe or because the oil was incorrectly selected or applied, are not being considered, since they are not of a fatigue nature. Also, cyclic stresses that develop from flexing of a bearing due to inadequate supporting members are outside the scope of this paper.

Fatigue eventually occurs in a bearing when cyclic stresses produced in the metal exceed a certain minimum value, which depends upon the composition of the metal and other factors, such as temperature, surface smoothness, etc., during the stress cycles. The requirement of cyclic stresses is found in all anti-friction bearings, because, when the rolling elements pass through the load area, each is subjected to applied stresses which are relieved as the ball or roller leaves that area. Anti-friction bearings are normally selected in any given application so that the stresses that occur are not sufficiently high to cause fatigue failure before a predetermined desirable period of operation has resulted.

A plain bearing that is subjected to a steady load that varies little in the direction or magnitude will not show fatigue failure. However, plain bearings of all reciprocating engines are subject to cyclic stresses, as well as plain bearings in other operations where cyclic shock loading exists. This should not be construed to mean that fatigue failures are common in plain bearings operating under these types of loads. Actually, in most industrial applications involving

* Sponsored by the Technical Committee on Bearings and Bearing Lubrication and presented at the 7th Annual Meeting, ASLE, Cleveland, April 8, 1952.

steam engines, air compressors, reciprocating pumps, etc., unit loading on plain bearings is normally below values that produce fatigue, and bearing failures from this cause are not common. The bearings of internal combustion engines, however, are designed for high loads; and small misalignment, poor fits and high local temperatures sometimes result in stresses that cause fatigue cracks and pitting of the metal.

In the analyses of the causes and means for preventing bearing failures, the effects of lubricant viscosity, film strength, rust preventive properties and ability to prevent ingress of contaminants into the bearing are subjects for consideration. A review of the role of the lubricant as related to bearing fatigue is discussed in this paper under these four subject headings. In view of the large amount of data that have been obtained in studies of fatigue failure of gears and the possible relationship thereof to bearing problems, several references are made in this discussion to the results of this work.

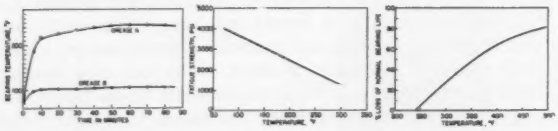


Fig. 1 (left) Effect of apparent viscosity on bearing temperature.

Fig. 2 (center) Fatigue strength of tin base babbitt vs. temperature.

Fig. 3 (right) Influence of temperature on ball bearing life.

EFFECT OF LUBRICANT VISCOSITY: Obviously, a bearing that is supporting dynamic loads sufficiently heavy to cause fatigue would quickly gall and seize if it were operated without lubrication. Under these conditions, it could not operate long enough for the metal to fail by fatigue. However, with the correct lubricant in service, the bearing could be employed without galling or excessive wear even though loads were sufficiently high to cause surface fatigue of the metal.

Under conditions of hydrodynamic lubrication, the lubricant, as a result of its viscosity and ability to adhere to metals, is capable of materially spreading the load, and reducing the maximum compressive stress in the bearing metal. For example, using an oil having a Saybolt viscosity of 185" @ 100 F., S. A. McKee and T. R. McKee[1] measured maximum oil pressure of 105 lbs./sq. inch in a bearing of 0.8753" in diameter and one inch long, which operated at 244 rpm under a load of 25 lbs. Under these conditions, the compressive stress in the metal failed to exceed 105 lbs./sq. inch. If the lubricant were not present, the journal under the same 25 lb. load (28.5 lbs. per sq. inch of projected bearing area) would produce a maximum Hertz compressive stress in the bearing of 890 lbs./sq. inch, or nearly nine times the stress produced when the bearing was lubricated with a full-fluid film.

In the above example, the bearing was lightly loaded. Under proper conditions of speed and oil viscosity, loads can readily be carried without danger of fatigue, but which, without the lubricant, would produce stresses beyond the endurance limit of the bearing metal. In fact, in properly lubricated bearings, such high loads can be carried on fluid films that the oil pressure developed in the film may be above the endurance limit of usual bearing materials. The loads on the bearings of many internal combustion engines, for example, have progressed beyond the fatigue limit of the softer alloys, and have led to the use of harder materials which have greater resistance to fatigue. In this connection, it is of interest to note that a modern automotive babbitt bearing often consists of a thin layer of babbitt (0.002 inches to 0.005 inches) which is chemically bonded to a steel back. The reduction from a relatively thick to a very thin babbitt facing has increased fatigue life of the bearing by many fold.

Since the endurance limit of metals generally decreases with an increase in temperature, the operating temperature of a bearing can be a factor that determines the load capacity of a bearing from the point of view of fatigue. An important role of the lubricant is to maintain bearings at reasonable temperatures. Two conditions, which are encountered in bearing installations, are (1) where high temperatures may result because of heat flowing to the bearing from an outside source, and (2) where excessive frictional heat is developed in the bearing itself. By circulating ample quantities of oil to bearings subjected to induced heat, normal bearing operating temperatures can be maintained. The front end bearings of many modern steam turbines, which are close to a zone through which steam at 1000 F. is flowing, are examples of bearings in which the lubricant must also function as a cooling medium.

It is well known that bearing friction, and hence temperature rise, is directly affected by the viscosity of the lubricant. Marked differences in bearing temperature rises can readily be demonstrated by employing, in plain or anti-friction bearings, greases having widely different apparent viscosities. Two such lubricants with apparent viscosities of 32 and 160 poises respectively at 100 F. and shear rates of 1000 reciprocal seconds have shown equilibrium bearing temperatures that differ by 150 F. when used in 80 mm-OD cartridge-type ball bearings, operating at 3400 rpm. Fig. 1 shows the actual operating temperatures in these bearings.

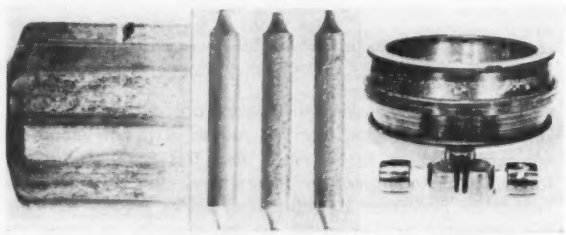


Fig. 4 (left) Section of rock drill piston showing cracked surfaces.

Fig. 5 (center) Rollers before and after needle bearing tests: (1) New roller, (2) Used roller—special lime base grease without EP additives, (3) Used roller—special lime base grease with EP additives.

Fig. 6 (right) Corrosion-fatigue in spherical roller bearing.

The extent of change of the fatigue limit of a bearing metal with an increase in temperature depends on the metal itself. Boegehold[2] has shown, for example, that the endurance limits of notched specimens of carburized gear steels are lower at 250 F. than at room temperature. Crankshaw[3] and MacNaughten[4] have also reported data which indicate a relatively rapid decrease in the fatigue limit of tin-base babbitt with increase in temperature (Fig. 2). On the basis of these data, the load that could be safely carried by a tin-base babbitt bearing operating at 130 F., from the viewpoint of fatigue endurance, would be reduced by about 60 per cent by increasing the bearing temperature to 300 F.

TABLE I—NEEDLE BEARING TEST

	Av. Bearing Temp. Rise Above Ambient F	Hrs. to Failure	Condition of Rollers just Prior to Failure
Special Lime-Base Grease without EP additives	60	400-600	Spalled areas
Special Lime-Base Grease with EP additives	63	400-600	Spalled areas

Likewise, the life of conventional type anti-friction bearings is reduced when high operating temperatures are involved. Although no specific data would appear to have been published indicating the change in anti-friction bearing fatigue life with an increase in temperature, Allen[5] has sug-

gested that it would be safe to assume that rated bearing capacity is permanently reduced by about 10% at 300 F., 25% at 400 F., and 40% at 500 F. On this basis, by applying these values to Manufacturers' Life or Safety Factors, a curve showing the effect of temperature on bearing life can be obtained, similar to that shown in Fig. 3. In this particular case, the curve was derived considering an 80 mm. single row angular contact light series ball bearing operating at 500 rpm. and under a combined radial and thrust load of 1403 lbs.

In addition, wide temperature changes can produce stresses within the metal of plain bearings that can cause fatigue. These stresses occur because of two separate effects: (1) as a result of the difference between the coefficient of thermal expansion of the bearing metal and the steel shell to which it is bonded, and (2) as a result of the thermal behavior of anisotropic solids.

With respect to the first of these effects, tests carried out by MacNaughton[4] show that stresses in the white metal after casting onto a steel shell and cooling to room temperature are found to be high and of the order of magnitude of the yield strength of the white metal. Upon remaining at room temperature, these stresses are reduced only by creep. If, however, the white-metalled shell is subsequently heated to 300 F., the stress in the metal soon drops to a small value, due to rapid relief of stress by a high rate of creep. On subsequent cooling to room temperature, the stress in the bearing metal regains its original high value. These tests seem to indicate that a cycle of operation and cooling down after shutdown subjects the metal at the interface to a considerable stress cycle, which, if of sufficient magnitude and frequency, could cause fatigue.

Frictional heat generated during operating may raise the temperature of local surface areas to quite high values, particularly if loads are heavy and lubrication is not adequate. Bowden[6] points out that in a polycrystalline specimen of an anisotropic metal, the orientation of any two neighboring grains will generally be dissimilar, so that when the metal is heated, the expansion on two sides of the grain boundary is different. The stresses set up are usually sufficient to produce plastic deformation and slip in the crystals, and the amount of deformation will increase with the number of cycles. Tin base babbitt has this characteristic to a marked extent and again if temperature variations are sufficiently great and frequent, thermal fatigue will result. To prevent fatigue due to these causes, the role of the lubricant as a coolant and as a lubricant cannot be minimized.

Surface fatigue failures sometimes occur which cannot be satisfactorily explained on the basis of design stress considerations, but which are due to the unit being subjected to imposed vibrations which are at its critical frequency. Under these conditions, high peak stresses may develop within the metal which exceed the endurance limit of the material. Gear-tooth failures from this cause have been observed in service installations and have been corrected by changing the frequency of vibration imposed on the unit. Changes in film strength or viscosity characteristics of the lubricant have shown no effect in retarding or preventing fatigue of this nature.

Reference should be made to the works of Buckingham[7] and Way[8], who have carried out extensive investigations on gear-tooth surface failures. Buckingham's tests have indicated that by straining the metal under high contact stresses, cracks will start below the surface of the metal, and subsequently extend to the surface and result in spalling. Way, however, reported only surface cracks due to stresses arising from surface irregularities, and pointed out that the lubricant entering these cracks produced a fluid pressure that caused the cracks to grow and eventually forced out pieces of metal leaving pits. Also, Way found that, in his particular test equipment, an increase in viscosity of the lubricant above a certain critical value effectively increased the load that could be carried without pitting. The viscosity required to effect this improvement, however, was higher than any practical value that could be used in the lubrication of enclosed gears.

Because of the many variables involved, fatigue data of this nature cannot be applied directly to bearings. However, there have been indications from field service experience that fatigue failures of diesel engine main and crankpin bearings ceased after the next heavier grade of oil was put in use. Furthermore, Hunt[9] has recently reported that endurance lives of main-rotor anti-friction bearings of British aviation gas turbines increase as the viscosity of the mineral oil is increased from 37 to 98 S.S.U. at 210 F. (3.15 to 20.0 centistrokes at 210 F.).

EFFECT OF FILM STRENGTH IMPROVING ADDITIVES IN THE LUBRICANT: As previously stated, operating bearing temperature levels can frequently be decreased through the use of correct-bodied lubricating oils or greases. Likewise, under conditions of heavy loads, where continuous hydrodynamic films cannot exist, equilibrium temperature levels of rubbing surfaces can sometimes be lowered through the use of so-called film-strength improving additives, which function by forming tenaciously held films on the load-carrying areas. In such cases, by preventing high-surface temperatures, thermal cracking and subsequent fatigue of the surfaces can, in some applications, be delayed or prevented.

An example that might be cited is encountered in rock-drill lubrication. Because of the very high speed reached by the piston of a modern drill as it travels back and forth two thousand times a minute, and the high load between the flutes of the rifle bar and piston and their rotating nuts, any failure of lubrication results in friction and local spin temfailure of lubrication results in friction and local skin temperatures that are extremely high. This high temperature causes thin surface layers to try to expand in all directions, but the relatively cold interior metal restrains expansion in any direction except directly outward. Thus, the metal is compressed beyond its elastic limit, and, on cooling is stressed in tension. If these tension stresses are sufficiently great, surface cracks will result that act as stress raisers and subsequently will lead to fatigue and premature failure. A view of the cracked surface of a rock-drill piston is shown in Fig. 4. It will be noted that a surface crack has extended, as a result of fatigue, and a large piece of metal has cracked out. When this condition is caused by insufficient film strength of the oil, it can be corrected by employing a suitable oil containing extreme-pressure type materials. In other words, through the use of an extreme-pressure lubricant, fatigue in this instance can be prevented by effecting lower temperatures of the rubbing surfaces.

Some lubricating oil additive materials, such as certain type phosphorus-containing compounds, have the capacity to "chemically polish" metal surfaces and thereby effect an improvement in surface smoothness. Such additives would appear to function by forming a low-melting eutectic with the surface asperities (minute peaks) which are polished away by the rubbing action. It is possible, therefore, in some instances to decrease surface roughness and effect better distribution of the load on bearing surfaces through the use of the proper type of additive material. Fatigue pitting due to surface roughness has been prevented by this means. In one such instance, moderately loaded, fairly large roller bearings, which had outer-bearing race surface roughnesses of 25-50 micro inches (rms), pitted as a result of fatigue in less than 1,000 hrs. when operating at 75-150 rpm and at a temperature of 275-300 F. The lubricant used was a non-EP type oil, having a Saybolt viscosity of about 600 seconds at

(Continued on page 314)

Lubrication Summarized

FUNDAMENTALS -OF -HYDRODYNAMIC LUBRICATION (Abstract)

by M. D. Hersey, U.S. Naval Engineering Experiment Station, presented at 1st National Symposium on Fundamentals of Friction and Lubrication in Engineering, ASLE, Chicago, September 8, 1952

Hydrodynamic lubrication means fluid film lubrication as distinguished from boundary, or "adsorbed film" lubrication. The first fundamental is to be sure to have enough lubrication fluid present to separate the surfaces. A second fundamental aims to have nothing but the lubricant present; avoiding aeration, and abrasive particles between the two surfaces. The size of grit particles limits the safe film thickness and load capacity.

A third requirement is to provide some kind of geometrical restriction, so as to build up pressure when the fluid is dragged into the restricted space by the moving surface. The restriction may be an abrupt step or dam, as in Rayleigh's bearing; a flat taper as in thrust bearings; or a curved, crescent taper as in loaded journal bearings. This pressure supports the load, and is distributed over the stationary surface in such a way that the resultant is equal and opposite to the load on the bearing.

Good adhesion is required between fluid and solid. If "slip" should occur, no pressure could be built up in front of the restriction. It is probable that all ordinary fluids adhere perfectly. The fluid must also be viscous, or it will immediately be squeezed out. Viscosity is necessary to maintain the pressure build-up and prevent contact. Without viscosity, the fluid could not be dragged along by the moving surface and forced against the restriction.

Viscosity causes frictional resistance, power loss, and heat generation. It should therefore be no greater than necessary to support the load. Viscosity is defined as the ratio of the applied shear stress to the rate of shear produced. Ordinary fluid lubricants follow Newton's law, according to which the rate of shear is directly proportional to the shear stress. By means of this law, both fluid friction and film thicknesses can be calculated in many applications.

Some lubricants, notably greases, do not follow Newton's law. Other examples are oils below their pour point, and oils solidified under high pressure. The laws governing shear stress and rate of shear in such materials are called "rheological laws," meaining laws of flow. In a perfectly plastic material there is no shearing deformation, or flow, until the shear stress exceeds a certain value. A live question of the day is to discover the limiting shear stress, if any, above which straight mineral oils may become "non-Newtonian."

The lubricant itself is the source of the frictional heat. Under hydrodynamic conditions this heat does not originate in the metal, or at the interface. Hence, the lubricant cannot act as a coolant in the ordinary sense. Outflow of lubricant is required in order to prevent accumulation of heat.

To initiate hydrodynamic action with minimum wear, motion may be started without load or by "hydrostatic lift." Professor Fuller has modernized and extensively applied the principles of hydrostatic lubrication, supporting the bearing loads continuously by pressure from an outside source. Lubrication may also be accomplished by thermodynamic action, as discovered by Fogg. The cooler fluid at the entering edge heats up during its passage through the film, developing pressure by its effort to expand. This pressure supports the load.

The viscosity of a lubricant is its most important physical property. Viscosity varies with pressure and temperature. Starting with a medium grade mineral oil at 210 F., the application of 10,000 pounds per square inch increases the viscosity three or four fold. This effect is equivalent to a drop in temperature of 40 to 50 degrees F. At still higher pressures, the effect is greatly intensified. Both Stanton and Needs have demonstrated the reality of the pressure-viscosity effect by journal bearing experiments. Needs and others have shown that the low-friction characteristic (or "oiliness") of fatty oils under heavy loads may be very largely explained by the pressure-viscosity effect.

In geometrically similar, steady-load bearings the coefficient of friction and relative film thickness depend primarily on ZN/P, the product of viscosity by speed divided by the load per unit area. It follows that the load capacity of a steady-load bearing, for any chosen film thickness, is proportional to the product ZN, where Z is evaluated at the effective temperature and pressure of the film. With rotating loads of constant magnitude and constant angular velocity, the same relation applies, provided the equivalent speed N is taken equal to the journal speed minus twice the load speed.

Professor H. W. Swift has set up the appropriate differential equations for dynamically loaded bearings in the general case, disregarding cavitation. A solution for Diesel engine connecting rod bearings was obtained by Professor Burwell at M.I.T., using the differential analyser. Further applications to engine bearings are very much in demand; and the underlying assumption that the clearance space is full of oil, with no cavitation, requires confirmation.

Other recent advances are mentioned, including the effects of temperature gradients across the film, local elastic deformation, and roughness. It is pointed out that water and air should both serve usefully as lubricants in high speed bearings. While hydrodynamic principles can be simply stated, their application requires critical study and discussion. Consideration must be given to misalinement, thermal expansion of metals, large scale deformation under heavy loads, and oil-film whirl under light loads. Experimental technique must be continually improved to make difficult measurements like film thickness and temperature distribution.

Hydrodynamic lubrication offers unlimited life to bearing surfaces operating under a steady load, and serves to reduce the stress concentrations under fluctuating loads. The future may therefore see a more conscious effort toward design factors and operating condtions that will promote this type of lubrication. The paper concludes with some reference to the larger sized thrust and journal bearings found in hydroelectric and marine service, and in rolling mills.

FRICTION AND WEAR OF GRAPHITE AND OTHER LAYER-LATTICED SOLIDS (Abstract)

by R. H. Savage, General Electric Research Laboratory, presented at 1st National Symposium on Fundamentals of Friction and Lubrication in Engineering, ASLE, Chicago, September 9, 1952

Experimental work on graphite sliding contacts of the past few years is briefly reviewed, and the frictional or wearing properties of several solids of laminar structure are compared, including graphite, molybdenum disulphide, boron nitride and mica. Of these, the graphite showed low friction in air but not in vacuum, and required, at low pressure, vapors or other kinds of adsorbed films for maintenance of lubricating surfaces. MoS_2 operated with low friction both in air and in the apparent absence of vapors, (i.e., in a vacuum of 0.05 micron); BN showed high friction and rapid wear in vacuum and in air, both moist and dry.

These diverse results strengthen the view that the laminar structure does not alone satisfy the lubrication requirement, while on the other hand the

(Continued on page 318)

Lubrication in the News

ANTI-SEIZE, the new Molybdenum Disulfide lubricant, is a stable, non-melting, multi-purpose lubricant with a phenomenal capacity to prevent seizing and galling at bearing pressures well over 100,000 lbs. per square inch; will lubricate at temperatures below sub-zero and up to 750 F. Bel-Ray's unique homogenizing process and the use of special synthetic additives makes it possible for Anti-Seize to remain in suspension ten times longer than dry molybdenum disulfide when mixed in machine and cutting oils. Available in 3 oz. tubes and 24 oz. cans. Write for illustrated bulletin: Bel-Ray Co., Inc., Green Village Rd., Madison, N. J.

BUNG BUSHING BARREL PUMP, a new pull-type pump for all around can filling use for light liquids, drops into drum opening and securely holds pump to drum. Fits snug around tube and prevents foreign matter from entering drum at this point; to operate: tighten thumb screw and lock both bung and pump with one operation; fits 1½" and 2" openings on 15 to 55 gal. drums; all steel except spout. For complete information write General Scientific Equipment Co., 2700 W. Huntington St., Philadelphia 32, Pa.

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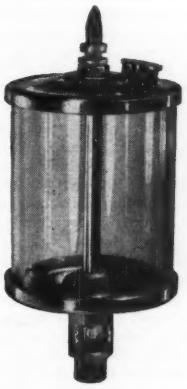
FERROFILTER, permanent magnet separator equipment, is designed for the effective removal of abrasive ferrous particles and chips from circulating lubricating systems on machine tools, internal combustion engines, plant and process equipment; from hydraulic oil lines on hydraulic presses and hydraulic systems of machine tool; from coolant systems on machining operations. Write for Bulletin 50 fully illustrated and including cross-sections to show how the units operate: S. G. Frantz Co., Inc., P.O. Box 1138, Trenton 6, N. J.

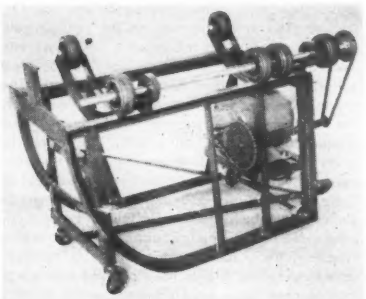
FORCE FEED LUBRICATORS CATALOG for all types of machinery gives full details on Hills-McCanna open and enclosed type lubricators which are available in capacities from 2 pints to 32 pints with 1 to 24 feeds per unit. Write for Catalog L-52, Hills-McCanna Co., 3025 N. Western Ave., Chicago 18, Ill.

GRACO BUS, TRUCK & FLEET CATALOG, No. 302, has been designed to show the operator how modern layout of the lubricating department with logically-arranged, time-saving equipment will help cut lube costs. Included in the catalog is the Graco Installation Planning Service, available to all fleet operators without charge. Write The Gray Co., Inc., 1014 Sibley St., Minneapolis 13, Minn.



GRAVITY FEED OILERS, newly developed large capacity aluminum alloy bodied, are similar in appearance and supplement to the recently improved line of Small Capacity Gravity Feed Oilers; dependable in performance, modern in design, they are extremely sturdy and yet light in weight and therefore especially suitable for locations where vibration is excessive. Drop feeding of oil is controlled by a precision needle valve which can be opened or closed by a knurled cap screw. Smooth easy hairline adjustments can be made because of a novel friction lock arrangement and any setting can be locked if needed against severe vibration. A shut-off toggle starts or stops oil flow but does not effect the adjustment. For further information on large capacity feed oilers write for Bulletin 61-11; for information on small capacity gravity feed oilers write for Bulletin 61-1. Oil-Rite Corp., 2328 Waldo Blvd., Manitowoc, Wisc.

GULF SDL—"ALL-YEAR AUTOMOBILE FUEL SYSTEM CONDITIONER" — recently developed to answer a widespread demand for such a product among motorists consists of a complex blend of ingredients selected to alleviate the common troubles encountered in fuel systems. It contains solvents to remove harmful gum which may be present in carburetor and fuel systems; it protects against further gum formation thus helping cleanse and keep important parts free-operating. In motor tune-up work, it aids in freeing sluggish or even lightly stuck valves; it is designed to aid in the lubrication of the upper cylinder region. As a drier, it absorbs moisture condensed in gas tank, fuel lines or carburetor thus providing an anti-freeze action for the fuel system; it helps to prevent engine stalling caused by formation of ice in the carburetor under certain atmospheric conditions. A corrosion inhibitor has been added with the aim of protecting carburetor parts against corrosion from moisture which may have condensed in the fuel system. The product is packaged in 12 oz. cans especially designed for ease of pouring into the gas tank. For further information write Gulf Oil Corp., Pittsburgh 30, Pa.



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MOLYKOTE—Type A, an effective lubricant for helping to overcome the energy waste in internal combustion engines and whose utilization of the peculiar characteristics of the unique molecular

(Continued on page 317)

Current Literature

ASTM BULLETIN, No. 184, Sept. 1952, *A Test for Aircraft Gear Lubricants*, by E. A. Ryder. There was published in the Oct. 1947 ASTM BULLETIN under the title *A Gear and Lubricant Tester* a description of the Pratt & Whitney Aircraft bench machine for the testing of gear materials and gear lubricants. Since that date, this machine has been put to use by several other laboratories and a large body of experience is now available. This machine offers a satisfactory method of assessing the anti-scuffing properties of gear lubricants, and its use is being called for in oil specifications. This article describes the current procedure for this purpose.

AUTOMOTIVE INDUSTRIES, Vol. 107, No. 4, Aug. 15, 1952, *Single Oil System for Engine and Torque Converter*, Anon. Description of Chrysler Corp's simplified fluid torque converter which eliminates separate oil pump and sump is given. The oil filter element is replaced every 5,000 miles. "The fact that the engine can draw upon a larger volume of oil is thought to actually lengthen the useful life of engine oil." Specified interval between oil changes is twice a year. With this drive 13 quarts of engine oil are used for the combined torque converter and engine system.

CHEMIE INGENIEUR TECHNIK, Vol. 24, No. 3, May 1952, *Technical and Economic Possibilities of Carbon Monoxide Hydrogenation*, by H. Tramm. The author reports on the considerable advances attained in this process. High activity iron catalysts have been developed; output and quality of the product has been improved. The process became economically more profitable. The production of lubricating oils from synthesis products is discusssed at length. Optimum viscosity behavior of oils is attained by polymerizing terminal straight-chain olefins (the number of C-atoms in the olefins being equal). High grade olefins could be obtained by using as polymerization feed stock olefins obtained by cracking cobalt-synthesis soft paraffin waxes. The iron-catalyst diesel-oil olefins have also the advantage that no cracking is required, because the primary products can be directly used as the feed stock. Synthetic lube oils offer particular advantages at very low temperatures. The "conveying limits," that is, the temperature at which the oils are still sucked up by the pumps, is very low. The ring-sticking properties are excellent, as well as the piston-corrosion values, much lower than those of fatty or non-fatty mineral oil lubes.

DIESEL POWER, Vol. 20, No. 8, Aug. 1952, *Which Lube Oil?*, Anon. The purpose of this article has been to outline the current status of various oils, particularly those of the additive type. It is intended primarily to discuss and explain the meanings of the various types of engine oils, their specifications and their expected performance. Attempt has been made to explain some of the mechanics of deposits, sludges and corrosion, as well as to point out some of the service conditions that require greater attention.

FUEL ABSTRACTS, Vol. 11, No. 6, June 1952, *Synthetic Oils*, by H. Kubbel & A. Meusel. Details of tests with synthetic lubricating oils developed during the war are given. The oils were synthesized by alkylating naphthalene, and appear to have the property of softening the hard carbon formed in diesel engines burning bunker oil. Engine tests with an oil equal to the best Pennsylvania grade and with the synthetic oil both having the same viscosity, showed that the latter gave a piston ring wear figure of 70.6 and a cylinder wear of 33.4, assuming the wear figure of conventional oils to be 100. The synthetic oil produced no ring sticking, but its viscosity increased slightly with use.

INDUSTRIAL LABORATORIES, Vol. 3, No. 6, June 1952, *Research Teamwork Produces Lubricants From Turpentine*, Anon. Naval Research Laboratory chemists had been working on a long-range basic research program in synthetic lubricants, sponsored by the Navy's Bureau of Aeronautics. By oxidizing turpentine's major constituent, alpha pinene, Dr. L. A. Goldblatt and fellow chemists in an Agriculture Department division had formed pinic acid. Naval researchers found that many excellent new classes of synthetic lubricants could be made by using pinic acid in combination with high-boiling alcohols. A plentiful substitute for adipic and sebacic acids was thus found.

INSTITUTE SPOKESMAN, Vol. 16, No. 4, July 1952, *Phthalocyanine Lubricating Greases*, by V. C. Fitzsimmons, R. L. Merker, and C. R. Singleterry. The increased operating temperatures possible for electric motors insulated with silicone resins has created a need for a grease which will lubricate ball bearings at 200 C. or above. The soap phase of ordinary or of silicone greases deteriorates within a very short time at this temperature. Phthalocyanine pigments have been shown to form greases from petroleum oils, silicones, aliphatic diesters, polyalkylene glycol ethers, and fluorocarbons. These greases contain 80% or more of liquid by volume and maintain their structure to above 300 C. They are water resistant, shear stable, and less susceptible to oxidation than corresponding soap greases. Phthalocyanine pigment is not abrasive. Phthalocyanine gelled silicone greases have lubricated ball bearings for 10,000 hours at 80 C. and for as long as 900 hours at 220 C. Special alloy steels are necessary in bearings operating at 200 C. or above. The behavior of the phthalocyanine greases in laboratory and in functional tests indicates that the high temperature limit for the successful operation of grease-lubricated ball bearings is now set by the chemical and thermal stability of the oils available for grease formulation and by the metallurgy of the bearings themselves. A bibliography is included.

Vol. 16, No. 5, Aug. 1952, *Quality Control of Lubricants*, by C. R. Burton. The lubrication control program now in use by the Hibbing-Chisholm district of the Oliver Iron Mining Co., was started by first analyzing the lubrication requirements of all equipment in use and comparing what was required with what was being used. It was first discovered that over sixty-five different oils and greases were used for winter and summer operation. By evaluating and consolidating, this number of lubricants was cut to 12 different products. This consolidation of lubricant brands and types has had a three-fold benefit: it minimizes the chance of the mixing of lubricants, it yields a benefit through volume purchasing of lubricants, and it requires less inventory and accounting expense. The second step in establishing the controlled lubrication program was to make sure that the people responsible for lubrication knew what lubricant was to be used and where. The third phase of the controlled lubrication program was to insure best lubrication at least expense. Reprinted by permission of Assoc. of Iron & Steel Engineers, from IRON & STEEL ENGINEER, Vol. 29, No. 5, May 1952.

JOURNAL OF COMMERCE, NEW YORK, Vol. 233, No. 17,727, Sept. 10, 1952, *Air-Dry Lubricant*, Anon. Able to be sprayed or brushed on, an air-dry lubricant has been marketed by the Electrofilm Corp., 7116 Laurel Canyon Blvd., North Hollywood, Calif. Called "Lubond," the product is described as containing effective lubrication characteristics at temperatures from minus 70 to 300 F. The product can be applied to steel, iron, stainless steel, lead foil, aluminum, brass, wood, cork, and other materials. It is also resistant to hydraulic fluids, gasoline, lubricating oils and water. The item is recommended for use on hinge pins, hydraulic cylinder, gears, threads, water valves, tracks, springs, floats, bolts and nuts, actuator screws, etc. Composition is not given.

JOURNAL OF THE INSTITUTE OF PETROLEUM, Vol. 38, No. 342, June 1952, *Investigations on the Determination of Kinematic Viscosity Using Institute of Petroleum Methods*, Anon. This report deals with an experimental investigation carried out by members of the Viscosity Panel of the Institute of Petroleum in their respective laboratories to provide data on the following subjects: Basing all viscometric measurements on a single value of viscosity as a primary standard, i.e., the agreed value for the viscosity of pure water at 20.00 C. Using master viscometers which could be based solely on the primary standard viscosity and which will not be subject either to kinetic-energy corrections or to surface-tension errors. The use of master viscometers and present Institute of Petroleum standard viscometers to evaluate oils in the viscosity range of 1 to 2000 cs by a stepping-up procedure. The determination of the repeatability and reproducibility of viscosity determinations at different levels of viscosity values.

Some Experiments on the Influence of a Solid Phase on the Oxidation of Low Viscosity Mineral Oil, by D. J. W. Kreulen. The presence of a solid phase affects both the length of the induction period and the rate of oxidation of a mineral oil. Besides glass, other solids such as sand, anthracite, retort carbon, and silica carbide are effective. The oil used in the experiment was a transformer oil.

OIL DAILY, Vol. I, No. 218, Aug. 11, 1952, *Navy Reveals New High Temperature Bearing Lubricant*, Anon. New high temperature lubricants, promising added efficiency in the operation of electric motors, have been developed in a five-year program in co-operation with the industries involved, the Navy Department [has announced]. By using the new high temperature lubricants which are now commercially available, electric motors can be operated continuously at 150 degrees centigrade (300 F.) and require relubrication at intervals of between 500 and 1,000 hours, the study indicates. Composition is not mentioned.

PETROLEUM ENGINEER, Vol. 24, No. 10, Sept. 1952, *Cutting Fluids for Machine Tools*, by C. M. Larson. This extensive discussion of cutting fluids covers the following topics: objectives of metal cutting; necessary properties of cutting fluids; types of cutting fluids, i.e., cutting oils (mineral oils and compounded) and soluble (or emulsifiable) oils; cutting fluid ingredients; factors in selecting proper fluids, e.g., type of cut, metal, tool used; articular uses of soluble oil emulsions; and soluble oil additives, including water softeners and anti-foam agent. There are 22 diagrams and illustrations, including a chart which gives cutting fluid recommendations for 9 types of cutting operations showing type and strength of compounding suggested for various tool materials, metals, and degrees of cutting.

PLANT, Vol. 6, No. 3, Sept. 1952, *Lube System, Bearings Subject to Accurate Calculated Design*, by R. J. S. Pigott. The author explains that the design of lubrication systems, including bearings, can be fairly accurately calculated, and he supplies information on major design factors, lubricant factors, industrial oils, engine lubrication, severity factor, foaming, viscosity index improvement, calculations, and even cutting oils. (This article is based on research report presented at Oil and Gas Power Div. ASME meeting, June 23-25, 1952.)

Section News

PITTSBURGH: Sept. meeting — Presentation of ASLE Charter by W. P. Youngclaus, Jr., ASLE Administrative Secretary, to Pittsburgh Section Chairman, A. A. Raimondi, Westinghouse Research Laboratories (See picture), and followed by a movie *The Manufacture and Care of Gears*, courtesy of Fellows Gear Shaper Co., and a symposium on "Gear Lubrication" with M. Miller as moderator. Future meetings: Dec. 19 and Jan 16, 6:30 P. M., Sheraton Hotel, Pittsburgh, Pa. (Submitted by F. R. Ross, Sec'y-Treas.)

BOSTON: Sept. meeting—D. F. Wilcock, General Electric Co., presented a paper entitled *Fundamentals of Ball Bearing Lubrication.* "Metal Working" was the subject of the Nov. meeting. Future meeting: Jan 19, 6:30 P.M., Smith House, Memorial Drive, Cambridge, Mass., subject "Re-refined Oil." (Submitted by A. S. McNeilly, Sec'y.)

BUFFALO: (See Directory for new officers.) Future meetings: Dec. 17 and Jan. 21, 6:30 P.M., Sheraton Hotel, Delaware Ave., Buffalo N. Y., (Submitted by W. H. Miller, Sec'y.)

CHICAGO: Oct. meeting—A "Stump the Experts" session with F. L. Gray, U. S. Steel Co., as moderator, and as panelists M. D. Gjerde, Standard Oil Co.; W. Albright, Shell Oil Co.; and W. Bryant, Magie Bros. R. Regan, Sun Oil Co.; B. Holmberg, Socony Vacuum Oil Co.; and J. Roznell, Texas Co., served as judges awarding a silver dollar to persons whose questions were not properly answered. Future meeting: Jan. 15, 6:30 P.M., Builders' Club, 228 N. LaSalle St., Chicago, Ill. (Submitted by A. S. Two, Sec'y-Treas.)

CINCINNATI: Sept. meeting — S. D. Craine, W. A. Jones Foundry & Machine Co., presented a paper entitled *Inspection and Interpretation of Gear Lubrication,* concerning gear tooth wear and failure, and stressing the fact that wear can be detected and corrected before gears are destroyed. Included, as follows, were the following types of gear wear and their descriptive remarks: Normal Wear—burnished satin appearance, Abrasive Wear — deep grooves perpendicular to the pitch line caused by foreign material, Scratching—early indication of abrasive wear, Overload Wear—rapid development of scratches from pitch line to tooth edge (first indication is heat and smoke), Ridging—forming of ridges which only take place in case-hardening hypoid gears and is a metallurgical condition, Rolling and Peening — actual plastic yielding of metal from vibration, Rippling — found only in case-hardened gears and giving the same indications as rolling and peening, Slight Scoring—slight scratches formed by high point on meshing gear (an E.P. oil may eliminate), Severe Scoring—caused by excessive loads and speeds, Initial Pits —forms at pitch line only and caused by high starting loads (these are inconsequential), Destructive Pitting—pitting spreads over tooth face, Spalling—large pieces of tooth face fall out due to metal failure, Corrosive Wear — caused usually when gears are at rest and is caused by the oil or whatever foreign matter is in the oil, Burning—metal turns blue at the gear tip and is caused by load and speed conditions, Interference—gouge at root of tooth caused by tooth mis-alignment, Grinding Checks—similar to scratching only having a definite pattern caused by faulty manufacture, Overload Breakage—large piece of tooth or entire tooth breaking out caused by impact load, Fatigue Breaking—large pieces of teeth breaking out caused by repeated shock, Cracking—caused by improper heat treatment of gear, Quenching Cracks — caused by improper heat treatment of gears. A.G.M.A. has published a pamphlet showing photographs of each type entitled *Gear Tooth Wear & Failure,* A.G.M.A. Print No. 11002. (Submitted by D. W. Moore, Section News Committee Chairman.)

CLEVELAND: Future meetings: Dec. 23 and Jan. 27, 6:00 P.M., Cleveland Engineering Society, 2136 E. 19th St., Cleveland, Ohio. (Submitted by R. H. Josephson, Sec'y.)

CONNECTICUT: May meeting — Election of Officers (See Directory), color movie entitled *Rust Control,* courtesy of the Fafnir Bearing Co., and speaker A. Ackerman, F. E. Anderson Oil Co., speaking on "Corrosion," with panelists C. T. Hewitt, Fafnir Bearing Co.; C. Gillette, New Departure Div.; E. Worden, Stanley Works; H. Fishbeck, Pratt & Whitney Aircraft; and R. F. Casella, Fafnir Bearing Co., as moderator. Oct. meeting — R. E. Bright, Dow Corning Corp., presented an address entitled *Silicone Lubricants* dealing with practical applications of silicone oils and greases in industry. (Submitted by A. F. Green, Chairman Ex-Officio.)

DAYTON: Sept. meeting—J. Robinson, Vickers, Inc., addressed the section on the subject *Industrial Hydraulic Equipmeent,* describing types of hydraulic pumps, particularly Vane pumps, and pointing out the advantages of balanced vs. unbalanced Vane pumps. Lantern slides illustrated unusual industrial equipment employing hydraulic application. (Submitted by A. H. Turner, Vice-Chairman.)

DETROIT: Sept. meeting—Annual stag golf outing open to friends as well as membership at the Hillcrest Country Club, with door prizes, entertainment, and sporting events. (Submitted by J. A. Brady, Treas.)

FORT WAYNE: Sept. meeting—Movie session entitled *Lubrication Ain't No Problem,* courtesy of Westinghouse, followed by soda pop and root beer. Oct. meeting—Family night featuring Dr. Llewellyn Heard and his "Combustion Magic" show, followed by coffee and sinkers. Future meetings: Dec. 11 and Jan. 8, 6:30 P.M., Keenan Hotel, Fort Wayne, Ind. (Submitted by J. H. Wright, Sec'y-Treas.)

KINGSPORT: Sept. meeting—H. E. Goddard, Blue Ridge Glass Corp, presented a talk entitled *Lubrication in a Glass Manufacturing Plant,* reviewing the different types of equipment involved in the process and progress made in the lubrication of this equipment. (Submitted by J. E. Fleenor, Sec'y-Treas.)

LOS ANGELES: Oct meeting—D. H. Moreton, Douglas Aircraft Co., Inc., discussed *A Review of Synthetic Lubricants,* covering their properties, the applications where each property excelled, their effects on materials, the general families of synthetics now in use, and their availability, cost and possible future, with the following conclusions: (1) Applications of synthetics primarily exist where operating conditions (temperatures, etc.) limit the use of petroleum products, (2) Costs are still high, but with increased production, will be reduced, (3) Future success will be greatly promoted by the correct application of synthetics, their indiscriminate use as "cure-alls" will tend to impair their future. (Submitted by R. C. Dishington, Sec'y-Treas.)

(Continued on page 318)

Patent Abstracts

Prepared by ANN BURCHICK from Official Gazette—Vol. 661, No. 4; Vol. 662, Nos. 1, 2, 3, 4, 5; Vol. 663, Nos. 1, 2, 3. Printed copies of patents are available from the Patent Office at twenty-five cents each. Address the Commissioner of Patents, Washington, D.C., for copies and for general information concerning patents.

LUBRICANT, Patent #2,609,342, J. R. White, O. H. Clark, and W. W. Woods, assignors to Socony-Vacuum Oil Co., Inc. A lubricant consisting essentially of a liquid selected from the group consisting of glycerine and ethylene glycol, containing from about 0.5% by weight and up to about 90% by weight of a comminuted solid selected from the group consisting of molybdenum disulfide, titanium disulfide, tin disulfide, zirconium disulfide and tungsten disulfide.

OXIDATION INHIBITOR FOR LUBRICATING OILS, Patent #2,609,343, C. E. Saunders and C. M. Murphy, Jr. A composition of matter comprising an alkyl diester of an organic dicarboxylic acid and an oxidation-inhibiting amount of a compound selected from the group consisting of phenothiazine, phenoselenazine, and phenotellurazine.

WAX LUBRICATING COMPOSITION, Patent #2,610,150, E. C. O'Neil, Anna L. O'Neil administrator of said E. C. O'Neil. A wax lubricating composition consisting essentially of one pound of lubricating wax selected from the group consisting of paraffin and ceresin, one-half pound of metal powder and a dispersing agent in the amount of one hundred thirty-six grains of tartaric acid crystals, said metal powder being dispersed throughout the composition.

NONCORROSIVE OIL COMPOSITIONS, Patent #2,610,151, J. H. Putman, J. Scott, and D. W. Ervine, assignors to Standard Oil Development Co. A non-corrosive oil composition comprising mineral oil of lubricating viscosity, 0.05 to 5% by weight, based on the total composition, of sodium benzoate, a solubilizer comprising 0.02 to 20% of oil soluble metal petroleum sulfonate, and about 0.05 to 5% of a stabilizer.

RUST INHIBITING COMPOSITION, Patent #2,610,919, M. P. Kleinholz, assignor to Sinclair Refining Co. A composition adapted to inhibit the rusting of iron and steel consisting essentially of a mineral oil as its base and about 1-2% calcium mahogany sulfonate and about 0.5-1% of a material of the group consisting of alpha-hydroxy lauric acid and the product resulting upon partial self-esterification of such acid and which contains in admixture the acid, the semi-lactide esterification product of the acid, and a relatively minor proportion of the lactide esterification product of the acid.

LITHIUM SOAP GREASE CONTAINING BASIC ALKALINE EARTH METAL SULFONATE, Patent #2,610,946, G. W. Eckert, assignor to The Texas Co. A lubricating grease composition comprising as the essential constituents an oleaginous liquid lubricating base as the major constituent, about 10-30% by weight of a lithium soap of a saturated soap-forming fatty material, and from 0.25 to 3% by weight of an oil-soluble basic alkaline earth metal salt of a sulfonated hydrocarbon wherein the weight ratio of alkaline earth metal to sulfur content is from 1.1 to 2 times that represented by the normal salt $M(SO_3R)_2$, where M is the alkaline earth metal and R is the hydrocarbon residue, said basic salt being in sufficient proportion to enable said grease composition to pass the copper corrosion test of U.S. Army specification 2-134.

LUBRICATING GREASE AND PROCESS OF MANUFACTURE, Patent #2,610,947, A. J. Morway, assignor to Standard Oil Development Co. The process of making a stable high temperature lubricating grease which comprises combining 2 to 20 parts by weight of fatty acid of the C_{12} to C_{24} range with 20 to 50 parts of an aromatic type mineral base lubricating oil of relatively high viscosity, adding 1 to 15 parts of organic material having a molecular weight below about 160 selected from the group consisting of carboxylic acids and aldehydes undergoing the Cannizzaro reaction and a sufficient amount of a metal base selected from the group consisting of alkali metal and alkaline earth metal bases to form a soap-salt complex with said fatty acid and said organic material, heating the aforesaid ingredients with stirring to a soap-cooking temperature for a period of time to substantially complete said soap-salt reaction, and thereafter gradually adding and blending in 25 to 50 parts of a relatively low viscosity and low volatility mineral lubricating oil of paraffinic type while applying heat, the amount of said paraffinic-type oil being not substantially smaller than that of said aromatic-type oil and substantially maintaining a cooking temperature between about 280 F. and 450 F. to form a grease of relatively high viscosity index and relatively low volatility.

LUBRICATING COMPOSITIONS, Patent #2,610,948, A. J. Morway and W. C. Howell, Jr., assignors to Standard Oil Development Co. A lubricating composition which comprises a synthetic lubricating oil having a viscosity at 210 F. within the range of from about 30 to about 60 Saybolt Universal seconds and a flash point above about 250 F. having combined therein from about 0.5% to 20.0% by weight of a viscosity index improver selected from the class consisting of polymers of acrylate esters, polymers of methacrylate esters, polymers of isobutylene and copolymers of styrene and isobutylene, said synthetic lubricating oil being a residue remaining after distillation at a temperature within a range of about 542 F. to about 740 F. of the mixture of products obtained by subjecting monoolefins to the action of carbon monoxide and hydrogen in the Oxo process.

OLEAGINOUS CORROSION PREVENTIVE COMPOSITION, Patent #2,611,711, B. L. Costello. An oleaginous composition characterized by the ability to prevent corrosion of metallic surfaces when applied thereto and to be easily removed therefrom, consisting of a homogeneous blend of about 47% by weight of soybean oil, about 33% by weight of sperm oil and about 20% by weight of linseed oil.

LUBRICATION COMPOSITION, Patent #2,611,742, E. M. Kipp, assignor to Aluminum Co. of America. A lubricating composition comprising a major proportion of a hydrocarbon lubricating oil blended with a minor proportion of a body consisting of animal and vegetable oils and fats and BF_3, said body being prepared by exposing said natural substance to direct contact with a compound selected from the class consisting of gaseous BF_3 and decomposable compounds yielding gaseous BF_3 and being present in such amount that the lubricating composition contains from 0.001 to 3% by weight of BF_3, said lubricating composition possessing extreme pressure characteristics superior to those exhibited by a similar composition in which the minor proportion aforesaid consists of said natural substance.

LUBRICATION COMPOSITION, Patent #2,611,743, E. M. Kipp, assignor to Aluminum Co. of America. A lubricating composition comprising a major proportion of a hydrocarbon lubricating oil blended with a minor proportion of a body consisting essentially of fatty acid containing from 4 to 21 carbon atoms and BF_3, said body being prepared by exposing the fatty acid to direct contact with a substance selected from the group composed of gaseous BF_3 and decomposable compounds yielding gaseous BF_3 and being present in such amount that the lubricating composition contains from 0.001 to 3% by weight of BF_3, said lubricating composition possessing extreme pressure characteristics superior to those exhibited by a similar composition in which the minor proportion aforesaid consists of fatty acid.

LUBRICATION COMPOSITION, Patent #2,611,744, E. M. Kipp, assignor to Aluminum Co. of America. A lubricating composition comprising a major proportion of a hydrocarbon lubricating oil blended with a minor proportion of a body consisting essentially of a substance selected from the class consisting of the fatty acid esters of monohydric alcohols and polyhydric alcohols and BF_3, said fatty acids being those containing from 4 to 18 carbon atoms, said body being prepared by exposing said fatty acid esters to direct contact with a substance selected from the class consisting of gaseous BF_3 and decomposable compounds yielding gaseous BF_3 and being present in such amount that the lubricating composition contains from 0.001 to 3% by weight of BF_3, said lubricating composition possessing extreme pressure characteristics superior to those exhibited by a similar composition in which the minor proportion aforesaid consists of fatty acid ester.

LUBRICATING COMPOSITION, Patent #2,611,745, E. M. Kipp, assignor to Aluminum Co. of America. A lubricating composition comprising a major proportion of a hydrocarbon lubricating oil blended with a minor proportion of a body consisting essentially of a substance selected from the class consisting of unmodified alcohols containing at least three carbon atoms, alcohol-ethers and ethers and BF_3, said body being prepared by exposing said substances to direct contact with a substance selected from the class consisting of gaseous BF_3 and decomposable compounds yielding gaseous BF_3 and being present in such amount that the lubricating composition contains from 0.001 to 3% by weight of BF_3, said lubricating composition possessing extreme pressure characteristics superior to those exhibited by a similar composition in which the minor proportion aforesaid consists of said substance selected from the class consisting of unmodified alcohols, alcohol-ether and ethers.

LUBRICATING COMPOSITION, Patent #2,611,746, E. M. Kipp, assignor to Aluminum Co. of America. A lubricating composition comprising a major proportion of a hydrocarbon lubricating oil blended with a minor proportion of a body consisting essentially of a substance selected from the class consisting of aliphatic, aromatic and heterocyclic mono- and polyamines, amino alcohols and amides and BF_3, said body being prepared by exposing said substances to direct contact with a substance selected from the class consisting of gaseous BF_3 and decomposable compounds yielding gaseous BF_3 and being present in such amount that the lubricating composition contains from 0.001 to 3% by weight of BF_3, said lubricating composition possessing extreme pressure characteristics superior to those exhibited by a similar composition in which the minor proportion aforesaid consists of said substances selected from the class consisting of amines, amino alcohols and amides.

LUBRICATING GREASE COMPOSITIONS, Patent #2,612,472, A. J. Morway and P. V. Smith, assignors to Standard Oil Development Co. A lubricating grease composition comprising a major proportion of lubricating oil thickened to a grease consistency with a combination of a

(Continued on page 319)

WEAR AND LUBRICATION OF PISTON RINGS AND CYLINDERS

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Book Reviews

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Application of Infra-Red Spectroscopy to Lubrication Engineering, by G. Rappaport 8-3 129
Automotive Drawing Compounds—Specifications and Applications by J. T. O'Reilly 8-1 14
Bacterial Decomposition of Soluble-Oil Emulsions, by W. Sonntag 8-5 234
Carbide Steel Milling With Cutting Fluids—A Progress Report, by J. R. Roubik 8-5 235
Coal Mine Lubrication, by C. W. Thompson 8-4 174
Development of a Turbo-Prop Synthetic Lubricant, The, by L. D. Christensen 8-4 177
Dispensing Characteristics of Lubricating Greases, by R. F. Strohecker 8-5 247
Evaluating Cutting Oils by Determining Their Effectiveness in Reducing Sub-Surface Grain Distortion, by W. E. Lauterbach & E. A. Ratzel 8-5 238
Filtration of Industrial and Lubricating Oils, by J. R. McCoy 8-2 66
Fretting Corrosion, by R. S. Barnett 8-4 186
Fundamental Knowledge of Lubricating Grease Structure, by B. W. Hotten 8-5 244
Grease Lubrication of Ball Bearings, by L. D. Cobb 8-3 120
Influence of Point of Application of Cutting Oil on Tool Life, by W. E. Lauterbach 8-3 135
Lubricant Classification, by C. A. Bailey 8-3 125
Lubricating Properties of Molybdenum Disulfide, by I-Ming Feng 8-6 285
Lubrication and the Load-Carrying Capacity of Gears, by E. T. Hutt 8-4 180
Lubrication in the Textile Industry, by K. M. Lowry 8-1 18
Lubrication of Knitting Machinery, by F. K. Fogleman 8-2 74
Material Specifications For Oil-Film Bearings, by E. B. Etchells 8-6 291
Metal Cutting Developments, by R. J. S. Pigott 8-6 289
NACA Studies of Mechanism of Freeting (Fretting Corrosion) and Principles of Mitigation, by D. Godfrey & E. E. Bisson 8-5 241
Oil Fog Lubrication, by D. G. Faust 8-4 183
Planned Lubrication as a Part of Plant Maintenance, by T. R. Witt 8-6 282
Role of the Lubricant in Bearing Fatigue, by E. S. Carmichael & R. B. Purdy 8-6 295
Steam Turbine Lubrication, by A. S. Morrow 8-1 10
Steel Mill Lubrication From Management's Point of View, by T. R. Moxley 8-5 232
Study of Rust Preventive Properties of Greases, A, by R. J. Burger, B. Rubin, & E. M. Glass 8-1 21
Tired Bearings—Reasons and Remedies, by J. Palsulich 8-6 293

Bailey, C. A.: *Lubricant Classification* 8-3 125
Barnett, R. S.: *Fretting Corrosion* 8-4 186
Bisson, E. E., & D. Godfrey (See Godfrey, D.) 8-5 241
Burger, R. J., & B. Rubin, E. M. Glass: *A Study of Rust Preventive Properties of Greases* 8-1 21
Carmichael, E. S., & R. B. Purdy: *Role of the Lubricant in Bearing Fatigue* 8-6 295
Christensen, L. D.: *The Development of a Turbo-Prop Synthetic Lubricant* 8-4 177
Cobb, L. D.: *Grease Lubrication of Ball Bearings* 8-3 120
Etchells, E. B.: *Material Specifications For Oil-Film Bearings* 8-6 291
Faust, D. G.: *Oil Fog Lubrication* 8-4 183
Feng, I-Ming: *Lubricating Properties of Molybdenum Disufide* 8-6 285
Fogleman, F. K.: *Lubrication of Knitting Machinery* 8-2 74
Glass, E. M., & R. J. Burger, B. Rubin (See Burger, R. J.) 8-1 21
Godfrey, D., & E. E. Bisson: *NACA Studies of Mechanism of Fretting (Fretting Corrosion) and Principles of Mitigation* 8-5 241
Hotten, B. W.: *Fundamental Knowledge o fLubricating Grease Structure* 8-5 244
Hutt, E. T.: *Lubrication and the Load-Carrying Capacity of Gears* 8-4 180
Lauterbach, W. E.: *Influence of Point of Application of Cutting Oil on Tool Life* 8-3 135
Lauterbach, W. E., & E. A. Ratzel: *Evaluating Cutting Oils by Determining Their Effectiveness in Reducing Sub-Surface Grain Distrotion* 8-5 238
Lowry, K. M.: *Lubrication in the Textile Industry* 8-1 18
McCoy, J. R.: *Filtration of Industrial and Lubricating Oils* 8-2 66
Morrow, A. S.: *Steam Turbine Lubrication* 8-1 10
Moxley, T. R.: *Steel Mill Lubrication From Management's Point of View* 8-5 232
O'Reilly, J. T.: *Automotive Drawing Compounds—Specifications and Applications* 8-1 14
Palsulich, J.: *Tired Bearings—Reasons and Remedies* 8-6 293
Pigott, R. J. S.: *Metal Cutting Developments* 8-6 289
Purdy, R. B., & E. S.: Carmichael (See Carmichael, E. S.) 8-6 295
Rappaport, G.: *Application of Infra-Red Spectroscopy to Lubrication Engineering* 8-3 129
Ratzel, E. A., & W. E. Lauterbach (See Lauterbach, W. E., & E. A. Ratzel) 8-5 238
Roubik, J. R.: *Carbide Steel Milling With Cutting Fluids—A Progress Report* 8-5 235
Rubin, B., & R. J. Burger, E. M. Glass (See Burger, R. J.) 8-1 21
Sonntag, W.: *Bacterial Decomposition of Soluble-Oil Emulsions* 8-5 234
Strohecker, R. F.: *Dispensing Characteristics of Lubricating Greases* 8-5 247
Thompson, C. W.: *Coal Mine Lubrication* 8-4 174
Witt, T. R.: *Planned Lubrication as a Part of Plant Maintenance* 8-6 282

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CONFERENCE HI-LITES: Papers presented at the Lubrication Engineering Conference sponsored jointly by the University of Pittsburgh, and the ASLE Pittsburgh Section, Nov. 20 & 21, University of Pittsburgh, included: *The Role of Lubrication in Industry*, by E. M. Kipp, Aluminum Co. of America; *Principles of Lubrication*, by J. Boyd, Westinghouse Research Laboratories; *Lubrication Principles as Applied to the Plain Bearing*, by A. A. Raimondi, Westinghouse Research Laboratories; *Properties of Lubricants*, by C. E. Trautman, Gulf Research & Development Co.; *Laboratory Testing of Lubricants*, by C. A. Bailey, U. S. Steel Co.; and *Lubrication Application Equipment*, by J. D. Lykins, Wheeling Steel Co. The conference concluded with a dinner meeting at the Hotel Sheraton, with C. M. Weckstein, Timken Roller Bearing Co., speaking on the subject *Roller Bearings*.

(Witt, Continued from page 284)

people how easily the highly polished surfaces of balls and races are damaged by these hard abrasive particles, it is at once realized also that ball bearings should be handled very carefully and kept clean at the installation. It was mentioned earlier in this discussion that we take every precaution to keep lubricants clean. To complete the job we found it necessary to set up very definite procedures, with the necessary equipment, for handling, cleaning, and installing ball bearings. While only ball bearings are mentioned, all mechanical work receives closer attention than was thought necessary a few years ago. Good mechanical work and good lubrication are the first steps in preventive maintenance and some of the procedures that were set up to accomplish this seemed a little absurd at first to some of the maintenance people. It is difficult to impress mechanics with the importance of some of these procedures and it requires constant follow-up to assure that they are being observed.

If compared with some of the other ways in which plant lubrication can be carried on, the reaction might be that the cost of this program would be difficult to justify. Of course, a fair return on the money spent is expected and this showed up after two years of work when the cost of lubrication was reduced approximately 20%. By listing all machinery and establishing an adequate frequency of relubrication, the work load for each lubrication man can be established. When this program had progressed far enough to lay out the work in a systematic sequence it was found that the number of men assigned to lubrication could be reduced and assigned to other duties.

There are other savings to be realized from good lubrication, one of which is reduced maintenance costs. There are thousands of anti-friction bearings in a large installation of textile machinery. On some of the elements of these machines there used to be an excessive number of ball bearing failures some of which, we have since learned, failed because the wrong type of lubricant was being used.

The maintenance and repair costs of the various elements of the machinery are collected on special maintenance and repair codes. It is known periodically what effect any changes we make in maintenance procedure, or in machine design, have on these maintenance costs. The maintenance costs on many of these elements has been substantially reduced either as a result of improved lubrication or a change in design which in most cases was brought about by a thorough study of lubrication.

There are many intangible results that it would be difficult to record. It is known, however, that there are fewer equipment failures and that bearing replacement has been substantially reduced. There is less spoilage of product from lubricants. The morale of the lubrication men is better and the maintenance people, including the foreman, are more interested in lubrication. They are more concerned about lubrication after each repair job than was true a few years ago. It is easier to keep the equipment and the floors clean because no excessive amount of lubricant is used and this, of course, helps to make the plant a safer place in which to work.

(Feng, Continued from page 288)

rubbing pressure, the greater the percentage of the oriented molybdenum disulfide.

5. SIMILAR EXPERIMENTAL RESULTS OF GRAPHITE: The orientation of the graphite surface film has been investigated by Jenkins[25], Campbell and Kozak[16], and Fullam and Savage[26]. Jenkins' electron diffraction experiments of graphite showed that, in polishing a graphite surface, the graphite laminae have a tendency to take up an oriented position with the basal plane parallel to the surface. Campbell and Kozak's electron diffraction picture of the carbon brush and the friction track on the copper plate

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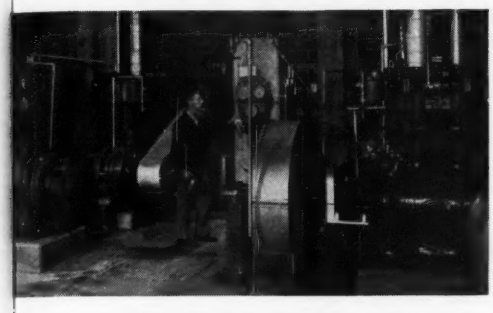

TEMPERATURES FROM −50F to +50F are supplied by these Frick ammonia compressors. They circulate brine through coils in the ice plant and under the arena floor, and ammonia through coils in the cooling and quick-freezing rooms. One grade of Suniso lubricates all of these compressors. Overall efficiency has been 15 percent better ever since its adoption.

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after running in the no-wear condition (in the presence of water vapor), showed oriented surface films on both surfaces with the cleavage planes in the plane of the plate and of the brush face. Their electron diffraction pictures taken after running in the wear condition (i.e. dry atmosphere) again showed carbon on both surfaces, but there was no sign of orientation.

Fullam and Savage investigated graphite surface films formed under the no-wear condition with the aid of the electron microscope. However, their result was quite different. The surface film was found to consist of many tilted "fingers", which were inclined at various angles up to nearly 90 degrees with reference to the plane of the rotating base.

SUMMARY: Molybdenum disulfide has a laminar structure, which is demonstrated by an electron micrograph of a molybdenum flake. The lubrication of molybdenum disulfide is independent of condensed vapors. This makes it more suitable for applications involving a vacuous atmosphere or one without a condensable vapor and also for applications at low temperatures.

In addition to independence of lubricating characteristics on an adsorbed film. molybdenum disulfiide is stable in air to temperatures as high as 400 C. These two properties combined together make it a good high temperature lubricant below its oxidation temperature, as well as a good high speed lubricant. Experimental results also show that molybdenum disulfide surface film gives a decreasing coefficient of friction with increasing normal loads and has a good resistance to penetration.

Friction of the cleavage surface of a molybdenum disulfide crystal has been found to be lower than that of the edges of the cleavage surface. X-ray diffraction study of a rubbed layer reveal an oriented molybdenum disulfide surface film in which the percentage of oriented laminae can be increased by increasing the rubbing pressure.

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(Pigott, Continued from page 290)

using the high-speed steel tools. Recent work, however, shows that we shall be able to provide new soluble oils that work well at 5% concentration.

Table II gives some results on the extremely troublesome Inconel M. Fig. 7 shows comparative life at varying speed with AISI 4147 steel. This shows the opportunity for increased tool life or increased speed, or a combination.

Smoke with the new system is definitely a problem when high speed and heavy cuts are employed. There are several ways of caring for smoke, all of them quite feasible. Ex-

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hauster systems, where available, will handle the smoke satisfactorily. For those cases where smoke is not freely dischargeable on account of air pollution, precipitrons do a first-class job of elimination. Next in order of efficiency come air-washers and certain types of filters.

But these methods are all fairly expensive, costing maybe more than the whole high-speed jet equipment. We therefore devised a very simple device that works well for all cases where it can be installed. A spray form was developed, using the former low-pressure circulating pump on the machine tool, with a ring of fine sprays mounted above the tool. This in many cases is enough; the spray head also serves to catch the high speed jet when not engaged in the cut. For severe smoke cases, a second spray ring is placed below the tool, shooting upward, with a horizontal spray closing the ring. Another form of spray head was developed employing a single slit spray instead of individual needle jets. The advantage of this latter form is that it is easy to make and the oil curtain is perfectly translucent. For very heavy cuts at high speed, this curtain contains the smoke formed at the tool, but the chips may be hot enough to continue smoking after breaking out of the curtain. For these cases additional quenching sprays must be used.

The standard Hi-Jet unit has a capacity of 1.4 g.p.m. at 400 p.s.i., and will supply eight 0.014 in. diameter jets or eleven 0.010 in. jets. With it will be supplied the suction and discharge gages, filters, relief valve and two lengths of flexible hose for connecting up. One nozzle setup will be provided with the unit. The cost, so far as can be definitely stated now, will be 2 to 4% of the cost of the machine tool to which the system is applied.

A little work has been done on other types of cut, so we feel confident the system can be applied to boring, milling, drilling, threading and grinding. Curiously enough, although grinding is very different from turning in many respects, we found early last year considerable improvement in thread grinding. But it will undoubtedly take quite some detail design to fit the jets to all applications. The equipment for turret lathes is already laid out and will be in use shortly.

It appears this system should be valuable on all long-run or production jobs where tool life is critical, or where increase of output is desirable. On the tough materials difficult to machine, it has produced improvement in tool life in every case. On the ordinary soft steels, it is possible to run at near carbide speeds with high-speed tools; for instance, SAE 1020 normalized, 400 s.f.m. cutting speed, 80 min. tool life.

Where tool life is increased by this method, tool inventory and tool grinding are reduced. A small quantity of the power units are available now and quantity production will be established shortly. Undoubtedly one or two sizes of larger units will be produced later, as we have already run into production jobs that require 10 g.p.m. pumps and 40 jets per machine.

New jet mounts have been designed, placed directly under the tool. These eliminate chip interference, do not require a separate location from the tool, and allow for applying the jets to boring and box tools.

There is a good deal of work yet to be done in detail on application, but it would seem this method offers an increase of production of the same order of magnitude as the increase of carbides over high-speed steels.

We did accomplish our original object—to get more consistent repeat tests; the Hi-Jet method varies far less than the conventional overhead system because build-up, a cause of uncontrolled variation, is eliminated.

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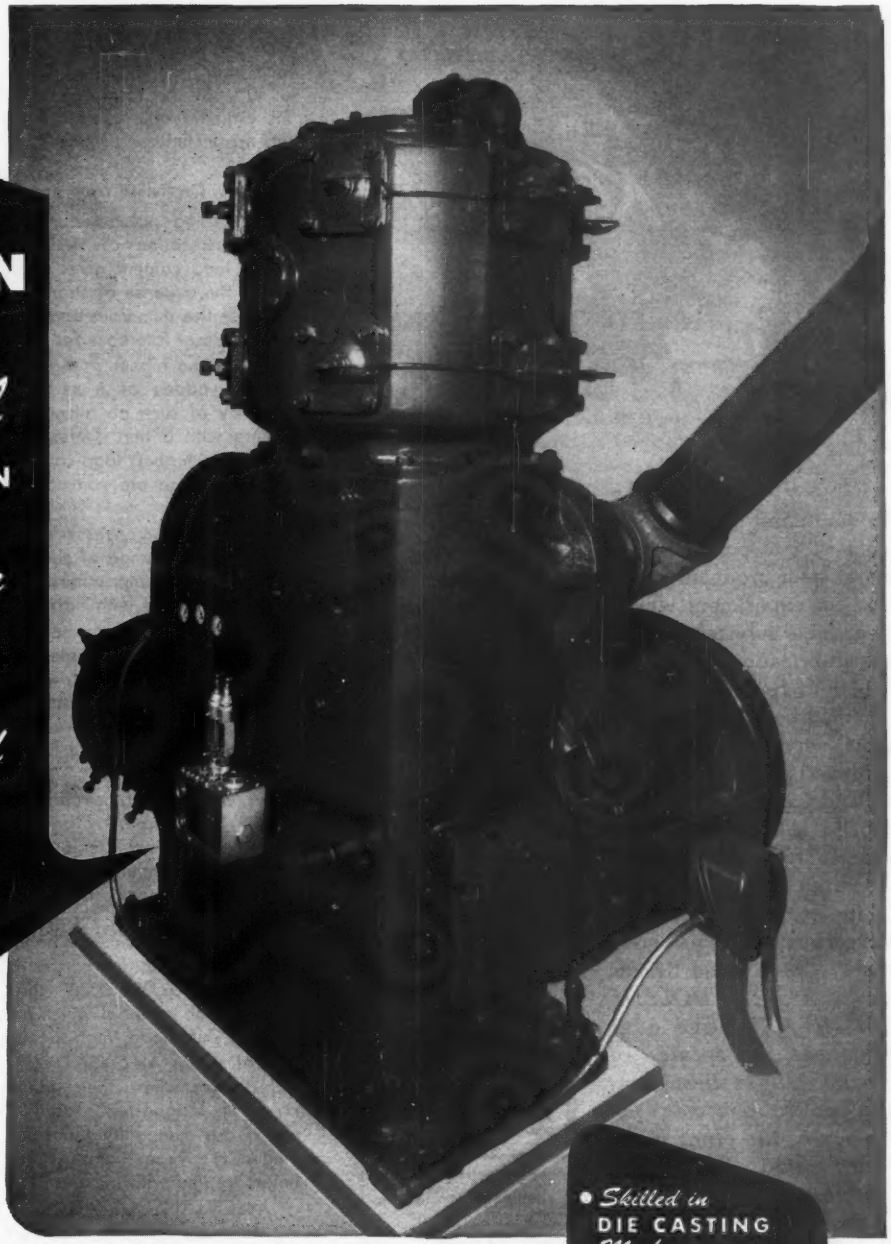

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To determine the relative performance characteristics of five competing open gear lubricants, a major steel producer established the 4-Gram Test . . . an experiment which reproduces as closely as possible service conditions encountered in open gear lubrication.

In this experiment the Timken Testing Procedure was modified slightly to reproduce more precisely actual service conditions. Flow of lubricant to the hardened and ground steel block and cup was cut off, and 4 grams of the lubricant being tested were applied at room temperature. The machine then was set in motion and watched closely for signs of scoring.

Five lubricants, all accepted by industry as "good quality," were tested. Results showed *that LEADOLENE KLINGFAST gave over 3½ times the service of the next best lubricant and up to 6 times the service of the other lubricants tested.*

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LUBRICANT "A"—5 minutes—a high grade compound scored at 5 minutes and failed in 12 minutes.

LUBRICANT "B"—8 minutes—a reputable lubricant of residual nature scored and failed in 8 minutes.

LUBRICANT "C"—6 minutes—a good lead compounded extreme pressure lubricant scored and failed in 6 minutes.

LUBRICANT "D"—6 minutes—a lead-compounded lubricant scored and failed in 6 minutes.

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A study of the inventory of tools installed and produced each year in the United States alone shows that from 30 to 40% of the totals could profitably use this system. In this estimate only tools on straight production or long-run cuts are included.

(Etchells, Continued from page 292)

process using powdered metals[12, 13]. Table VII gives the SAE specification of standard commercial analysis. The higher lead content gives a higher anti-score characteristic, but at the expense of strength. SAE 480 has a broad range to cover the numerous compositions; so the designer should specify closer tolerance for purchasing.

To obtain higher strength, tin up to 3% and silver up to 5% are added at a sacrifice of anti-score quality. The tendency of such an alloy to score can be offset by overplating with a thin coating of lead alloy, which results in one of the highest load-carrying plain bearings now in modern use. These are commonly known as tri-metal materials (Fig. 4).

BRONZES: Bronzes are valuable because of their low cost and wide range of characteristics as determined by the many various compositions. They are especially resistant to wear when used with hard shafts or journals.

The number of commercial bronzes is very large. The SAE specifications in Table VIII can be used as a guide in conjunction with Table I, which gives typical examples for their use. Numerous variations of these bronzes and special purpose bronze alloys are to be found in the ASTM Standards, Part 1-B, and in the SAE Handbook.

ALUMINUM ALLOYS: There are three aluminum bearing alloys commercially available as shown in Table IX. The XB-80S is a strip alloy and may be obtained with a steel backing, whereas 750 is furnished without steel backing. Bearing manufacturers should be consulted for specific information on the best application for these compositions. The 3889M material recently announced[14] is produced by the strip process, securely bonded to a steel backing (Fig. 5). The finished bearing carries a lead-babbitt overplate approximately .0003"-.0008" thick. This is another tri-metal type bearing material which may be classified in the moderate cost, high-load, high-speed group.

CADMIUM ALLOYS: These alloys are used in very special designs where their added cost is justified by the increase in fatigue life. Table X gives the SAE specifications which are essentially the same as may be obtained commercially. These materials are attacked by corrosive oils unless treated with indium.

MISCELLANEOUS ALLOYS: Two special types of bearing materials that are important commercially but restricted to specific duties are zinc castings and sintered powdered metal, the porous types.

Zinc castings are used where low cost is desired and where the load and speed are low. Zinc and its alloys are not good anti-friction metals and will score readily if the load and speed are too high.

Sintered powdered metal bearings are important for locations where oil is rarely applied. These bearings are referred to as "oil-less," but they function because oil is retained in the pores of the sintered metal. Typical service is in washing machines and small electric motors.

CONCLUSIONS: The determination of the material specification for a plain bearing application involves a compromise of the various factors which have been discussed. In the functional chart, (Fig. 6), the four important requirements are shown[3] for the most popular materials. Because it is very difficult to assign exact values to these properties, the length of each line may be taken as a measure of approximate order.

The selection of the proper bearing material for any application is an art in which competent experience plays an important part. When new problems and new conditions arise, it may be necessary for even the most experienced person to obtain new data.

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(Palsulich, Continued from page 294)

in several ways. Experience has shown that "worn-in" gears seldom create erosion fatigue damage in bearings. Therefore, any means to cause early wear-in of gear teeth will be useful. Extreme pressure lubricants have been found quite effective. A chemical phosphide treatment of the gear teeth has been found to be equally effective. In each of these cases the gear mating action is smoothed up by rapid wear-in of the teeth. It should be mentioned that in each of these instances, erosion damage will occur on the bearings during this run-in period, but that no damage will occur on replacement bearings. A method used successfully in many instances has been to alter the gear tooth profile in a manner which results in smooth mating action.

BEARING DESIGN CONSIDERATIONS: (1) MATERIALS AND CONSTRUCTION: Recognizing the fact that the selection of a bearing material for a given application is usually a compromise, the final choice must be dictated by the over-riding requirement. Perhaps cost may be the deciding factor. In any event, an array of materials is available from which the optimum functional choice can be made.

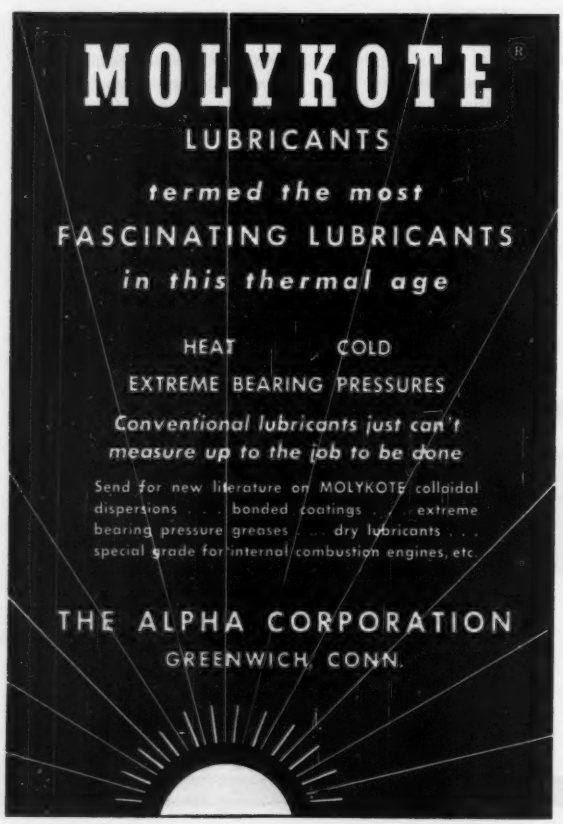
The proper ratio of lining-to-backing thickness has been established by experience, and the information is made readily available to the designer by the bearing manufacturer. The constructions described earlier in this report are common to the industry.

(2) TEMPERATURE: The ever-increasing bearing loads and speeds make it mandatory that particular attention be paid in design to keep the bearing operating temperature at a safe value. The maximum permissible operating temperature must be established by the bearing designer (who is aware of the limitations of his materials) and *not* by the engine builder or operator. The deleterious effect of temperature on bearing fatigue life is shown in Fig. 1 for babbitt bearings.

While the same curve may not be directly applicable to the copper-leads, bronzes, aluminum and silver, the effect of temperature on the overlay materials which are quite often used on the above is quit the same! For this reason, and to avoid too rapid oxidation of the lubricant and possibility of bearing corrosion, the upper limits of safe bearing operating temperature are set at 275-300 F., depending upon the material used.

Several steps must be taken to insure that the safe bearing operating temperature will be maintained, as follows: (a) Provision of adequate bearing clearance at all operating and ambient temperatures to which the bearing assembly may be subjected, keeping in mind the possible differential expansion rates of bearing, housing and journal. The effect of bearing clearance on oil flow (and temperature) is very significant and is often the factor which decides whether or not the bearing functions in a satisfactory manner. (b) Provision of positive escape passages to permit the free flow of lubricant from both ends of the bearing. This quite frequently necessitates grooving the thrust faces of the bearing or washer to avoid sealing off the flow due to either direct or indeterminate thrust face loading. (c) In the operation of automotive engines it is necessary to have very small bearing clearances due to the fact that, as vertical diametral clearances increase over .0022", there is also an objectionable increase in bearing noise. The oil flow through a bearing can be increased in various ways when small clearances must be maintained. Among these is the use of an eccentric bearing where the vertical clearance remains small so as not to increase bearing noise, but the horizontal clearance is increased. In one specific case, an eccentric bearing had a vertical diametral clearance of .0009" and a .0017" horizontal diametral clearance. In this design the oil flow at 2000 r.p.m. increased to .020 gal./min. from the .009 gal./min. for the concentric bearing. Under these same conditions the temperature versus speed relationship has been calculated and plotted as shown in Fig. 4. Note that for the concentric bearing under minimum clearance conditions of .009", the oil flowing from the bearing is above 400 F. at 2000 r.p.m. The eccentric bearing with .009" vertical and .0017" horizontal clearance reduces the oil spill temperature at 270 F. at the same speed. As stated previously, high temperatures caused by low clearances and insufficient oil flow have several bad effects on bearing performance, among these being the rapid reduction in fatigue life of the bearing material. Oil spreader grooves, circumferential grooves and parting line chamfers are also frequently used to increase oil flow. (d) The provision for adequate lubrication under all conditions of operation cannot be over-emphasized. Particular reference is made here to the condition wherein the engine is operated for short but dangerous periods of time without oil pressure. This often occurs on the start. The 30 to 60 seconds of oil starvation quite frequently results in initial damage to the bearing and localized fatigue. In many cases, however, it starts the vicious circle of bearing and shaft scoring which eventually

100 F. Under these conditions of operation, pitting occurred may lead to complete bearing seizure. This has been the author's experience with aircraft engines, and the same situation may exist in automotive and diesel installations. There is some reason for belief that this is one of the most frequent (but very often unrecognized) causes of sleeve bearing failures today. With present engine designs, engine pre-oiling is the best remedy. (e) In applications wherein the bearings are urged to carry extremely high loads, the shaft deflections may be great enough to severely end load and fatigue the bearings. This has occurred in high-powered aircraft engine master rod bearings. A solution to this problem was found by end relieving the bearing I.D. to a depth of approximately .001" for lengths from 1/4 to 3/8". It has been found more practical to determnie the amount of relief in each case by empirical methods rather than by calculation. (f) In specialized cases where extreme lightness of shaft and housing are necessary but the loads are high, the resulting deflections of the parts may result in a condition which the bearing designer can alleviate only by complicated and costly measures. Aircraft engine high speed planetary reduction gear systems are a case in mind. The situation was severe in reciprocating engines but is doubly so in turboprop gas turbines wherein tremendous power must be transmitted by the reduction gear system to the propeller. Unit bearing loads of 2000 p.s.i. at rubbing velocities of 70 ft./sec. must be carried on oil whose viscosity is only slightly better than that of kerosene. Besides this, there is no assurance that the total load is equally divided between the several planet pinion bearings used. In this instance, a successful bearing design has been evolved in which an extra oil film is provided on the O.D. with provision made to prevent rtation. This is done either by the use of splines at one end or end slots engaging mating stationary members.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It is hoped that an idea or two presented in this paper will furnish a clue toward the solution of some existing bearing fatigue problems; but more so, that known facts on sleeve bearings be judiciously applied in the design stage so that there will be fewer and fewer "tired" bearings.

(Carmichael-Purdy, Continued from page 297)

in a relatively short time. By employing a higher film strength lubricant, of the type referred to above, pitting was eliminated and bearing life was extended to that of normal duration.

It might also be pointed out that the pitting of gear teeth at the pitch line, where only rolling action is involved, can often be prevented by running in the gears for a short time under overload with an extreme-pressure lubricant. It is believed that the mechanism of action in this case is similar to that discussed above, namely, a smoothing of the surface in the pitch-line region with a subsequent improvement in load distribution before fatigue failure can occur.

It should not be construed from the above that all anti-friction bearing fatigue failures are due to surface roughness or that an extreme-pressure lubricant can usually prevent or retard fatigue. The results of evaluation in a needle-bearing tester, using a special type lime-base grease with and without extreme-pressure additives (lead soap and sulfurized fat), illustrate this point. The needle bearings (0.75" bore, 1.0" OD and 0.75" long) of this tester were operated at a speed of 1,000 rpm and under a load of 865 pounds per bearing until failure occurred. Bearing temperatures were recorded during the runs, and the rollers were examined prior to complete bearing failure. The results of these tests are given in Table I.

Because the surfaces of these rollers were finished to a high order of smoothness, and because of their spalled appearance, as shown in Fig. 5, it must be concluded that the entire surface was subjected to stresses beyond the endurance limit of the metal. It is evident from the results that, in this case, the extreme-pressure additives had no effect either in retarding or preventing fatigue.

EFFECT OF RUST-PREVENTATIVE PROPERTIES OF THE LUBRICANT: A large amount of experimental work has been carried out and reported on the deleterious effects of corrosion on the fatigue resistance of metals. McAdam[10] has shown, for example, that cyclic stresses accompanied by corrosion is considerably more harmful from the fatigue viewpoint than cyclic stresses without the corrosion. Since the nucleus for fatigue need be only a minute spot on the surface of the metal, it is obvious that corrosion can lower the endurance limits at which the surfaces will deteriorate.

The polished surfaces of anti-friction bearings are particularly susceptible to rusting whenever water enters the bearing, either as a result of condensation or leakage. One of the important functions of the lubricant is to protect these surfaces against corrosion.

Lubricating oils and greases differ materially in their ability to protect against rusting, and some products contain addition agents specifically designed to inhibit rusting. The effectiveness of rust preventative additives depends not only upon the chemical composition thereof but also upon the quantity of water present and its corrosive characteristics. To the extent that the lubricant can prevent corrosion under any given set of conditions, to that same extent will the bearing be protected against corrosion-fatigue.

It is not always possible to prevent fatigue of this type through choice of the lubricant. In a recent case, for example, corrosion-fatigue developed in spherical roller bearings, as shown in Fig. 6, because of leakage of river water into the bearings. The ingress of water could not be prevented, but by changing from the use of a highly corrosive

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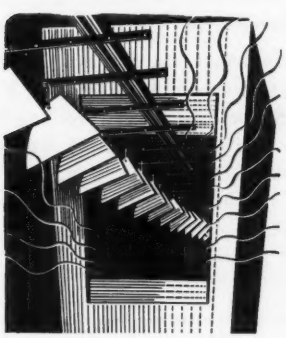
Bearings working at 375°F. in an oven conveyor system for baking the finish on electrical parts, failed frequently in spite of weekly relubrication. And many finished parts had to be rejected because of flaws caused by oil bleeding out of the bearings and dripping onto the uncured paint. Since using Dow Corning 41 Grease, bearing failure and rejects caused by melting grease have been eliminated, and the bearings have to be relubricated only twice a year.

After more than 30 months of service, conveyor bearings lubricated with Dow Corning 41 Grease are still running smoothly through an enamel drying oven operating at 300°F.

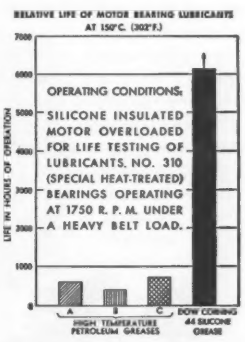


Bearing failures have been eliminated, maintenance and lubrication costs have been reduced to the vanishing point; production has been increased by at least the $100 worth of parts a day that were formerly rejected because the finish was spoiled by the dried-up, graphite-kerosene lubricant that fell down from the bearings.

As an antiseize agent on the die threads of glass forming machines, Dow Corning 41 Grease makes it easy to take dies apart even after 6 months of operation at 340-380°F. Similarly, die assemblies used in molding phenolic closures formerly froze so tight that a welding torch was used to cut the dies and bolts apart. The same assemblies lubricated with Dow Corning 41 are easy to disassemble after long exposure to heat and steam. Hold-down bolts on rubber and plastic presses are also lubricated with Dow Corning 41 to prevent seizure.

DOW CORNING GREASE 44

Count the number of conventional motors you have in your plant. Multiply that number by $4.52 and you have the average yearly cost of lubricating these motors plus the cost of motor failure caused by improper lubrication. With an estimated 10 to 15 million motors now in service, American industry spends 45 to 67 million dollars a year that might be saved by the use of Dow Corning Silicone Grease 44. That possibility is based on the performance of this Dow Corning Silicone grease in about half a million cartridge ball bearings used in a widely advertised line of industrial motors. Experience has proved that these bearings, prelubricated for life with Dow Corning 44, require no further attention.



Annealing Ovens at 1400°F are located only 2 feet from bearings on motor directly coupled to forced draft fans. In spite of frequent relubrication with organic greases these bearings had a service life of about 6 months. The electrical foreman finally installed bearings prelubricated with Dow Corning Grease 44. These bearings have never been relubricated and they are still in service after about 2 years of operation.

Such performance confirms the testing done by Dow Corning, by bearing manufacturers and by the military establishment. It indicates that you can expect Dow Corning Silicone Grease 44 to last at least 10 times longer than high temperature petroleum grease.

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river water to water having lower chloride and sulphate contents, a lubricant was selected that protected against corrosion-fatigue.

It should be pointed out that corroded areas on bearing surfaces may be worn away before cyclic stresses can cause fatigue. However, if the corroded area is not worn off, even though minute in size, it can act as a stress raiser and eventually lead to fatigue, if cyclic stresses are above the endurance limit.

EFFECT OF SEALING PROPERTIES OF THE LUBRICANT: In some applications where inadequate bearing seals are employed, and particularly where greases are used, the lubricant can sometimes play an important part in preventing contaminants, such as solids, corrosive liquids, etc., from entering the bearing. By preventing entrance of foreign materials, the lubricant can, in some cases, retard or prevent fatigue failures which might otherwise have resulted from contamination.

Solid contaminants that migrate between race and rolling elements of an anti-friction bearing can, if sufficiently hard, cause dents in the race track. The edges of such dents tend to be highly stressed by the rolling elements and crumble rather quickly in fatigue. In some cases, a small dent may grow larger and larger until the bearing fails, whereas in other cases, the rolling elements will either roll or wear the edges of a dent smooth before fatigue can occur. Dents or scratch marks in radial babbitt bearings generally will not result in fatigue failure.

Greases differ widely with respect to their ability to seal out extraneous materials from bearings. Consistency and resistance against structural breakdown under shearing action are two of the characteristics upon which this ability depends. Although some greases are quite effective in preventing the entrance of contaminants, this property cannot be relied on as a substitute for properly designed bearing seals.

SUMMARY: The possible effects of certain of the characteristics of lubricants on bearing fatigue, including viscosity, film strength, corrosion prevention and ability to seal against contaminants, have been considered in the light of present day knowledge. Generally speaking, bearing fatigue failures are not directly influenced by the lubricant. However, in some instances, the aforementioned characteristics of oils and greases may play an indirect role in minimizing this type of bearing surface deterioration.

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(A Memo from the Editor)

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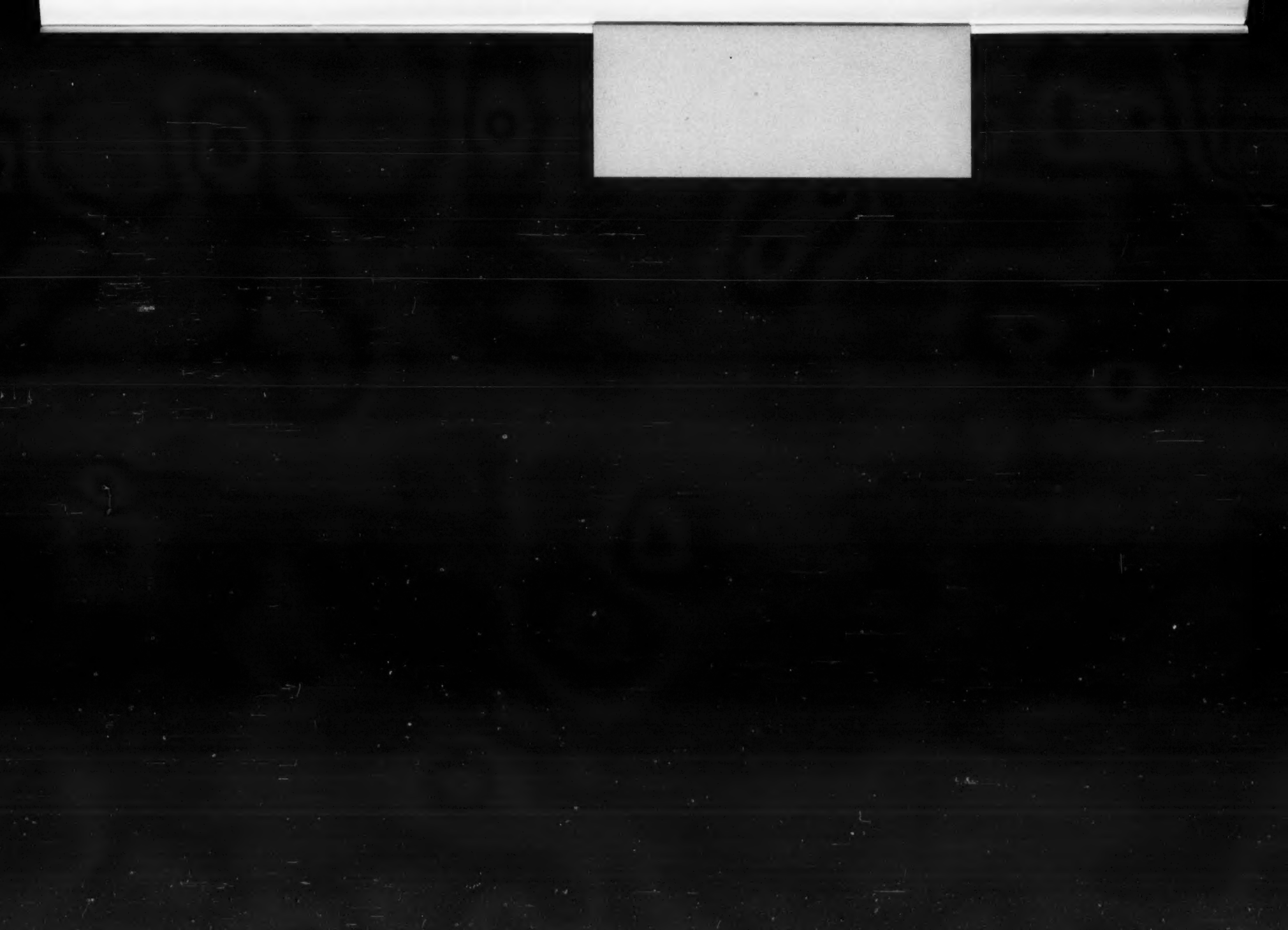

Coming Events

DECEMBER

26-31 American Association for the Advancement of Science, Jefferson Hotel, St. Louis, Mo.

JANUARY

7-8 Kentucky Petroleum Marketers Association (Annual Meeting—Trade Show), Brown Hotel, Louisville, Ky.

12-16 Society of Automotive Engineers (Annual Meeting & Engineering Display), Sheraton-Cadillac, Detroit, Mich.

15-16 National Industrial Conference Board, Astor Hotel, New York, N. Y.

19-23 American Institute of Electrical Engineers (Winter General Meeting), Hotel Statler, New York, N. Y.

26-27 Compressed Gas Association, Inc. (Annual Convention), Waldorf-Astoria, New York, N. Y.

26-29 American Society of Heating & Ventilating Engineers, Conrad Hilton Hotel, Chicago, Ill.

26-30 Eleventh International Heating & Ventilating Exposition, International Amphitheatre, Chicago, Ill.

FEBRUARY

1-6 American Society for Testing Materials (Com. D-2 on Petroleum Products & Lubricants), Hotel Cleveland, Cleveland, Ohio.

5-7 Missouri Petroleum Association (Annual Convention), Hotel President, Kansas City, Mo.

APRIL

13-15 American Society of Lubrication Engineers (8th Annual Meeting & Exhibit), Hotel Statler, Boston, Mass.

(Lube In the News, from 299)

structure of molybdenum and sulfur which makes it a natural lubricant, is described in Bulletin 96 and offered by The Alpha Corp., 179 Hamilton Ave., Greenwich, Conn.

TRABON CENTRALIZED LUBRICATION SYSTEMS, a 4-page two-color illustrated folder, covers pertinent sales points about the application of patented Trabon automatic lubrication systems, and includes engineering description of technical methods of operation in the case of Trabon's Reversible and Manifold types of lubrication systems. Write for Bulletin 529, Trabon Engineering Corp., 1814 E. 40th St., Cleveland, Ohio.

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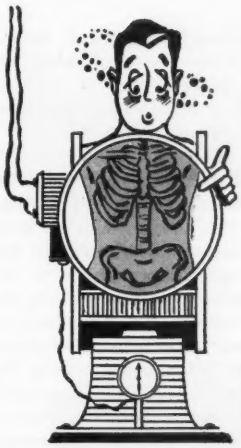

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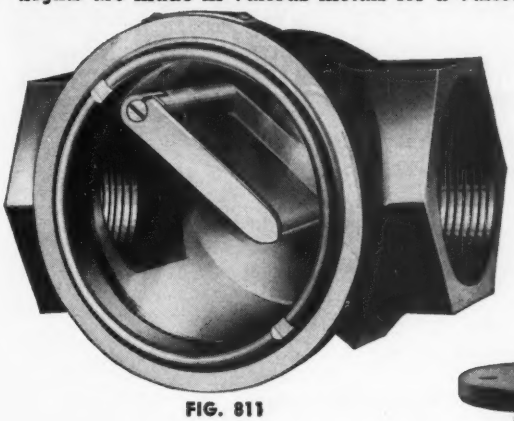

FIG. 811

This vane-type Bowser Indicator is spring-actuated, thus providing an indication of the quantity flowing in the line.

FIG. 811

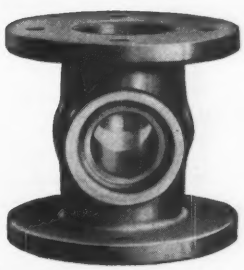

FIG. 54A

This Bowser Flow Sight is recommended for use on gravity lines where the flow is vertically downward.

FIG. 54A

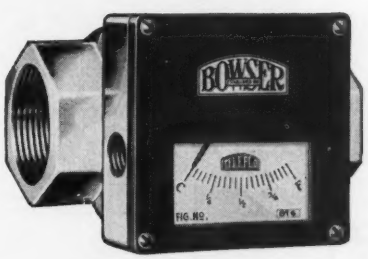

FIG. 816

The Figure 816 Bowser Telefio Indicator is equipped to operate a gong or to stop pump motor if liquid flow stops.

FIG. 816

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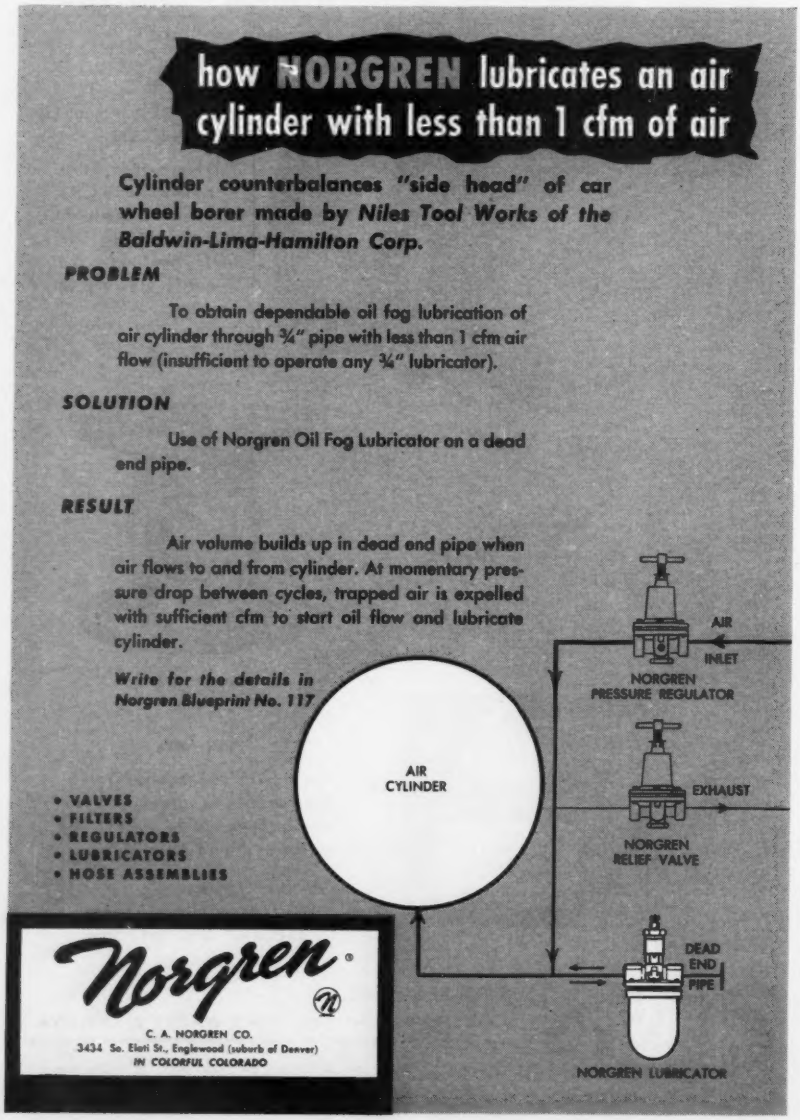
(Lube Summarized, from 298)

very low friction reported for diamond, in air, shows that a substance may be well lubricated by adsorbed films alone even though possessing a very unfavorable (tetrahedral) structure. A conclusion which follows is that while the structure undoubtedly has at least an important effect in promoting deposition of relatively impenetrable thin smooth solid films on bearing surfaces (cf. graphite and diamond), the primary requirement for completing the lubrication is that the surface fields of force must be compensated. With carbon this occurs through adsorption, and the powerful gettering properties of the wear dust indicate the magnitude of some of these forces when lubrication is incomplete. With boron nitride the lubrication was invariably incomplete and the wear dust did not show strong adsorption. It has not been determined whether the MoS_2 when operating smoothly in vacuum does so by virtue of truly intrinsic lubricant characteristic (e.g., weak S-S bonding) or whether some hitherto unsuspected surface films play a part.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(Section News, from 301)

MILWAUKEE: Sept. meeting — R. O. Bartels, Socony-Vacuum Oil Co., presented a talk entitled *Some Economical Aspects in the Storage, Handling, and Dispensing of Petroleum Products.* Oct. meeting—Tour of Norberg Mfg. Co. plant. Future meetings: Dec. 15 and Jan. 19, 6:30 P.M. E.S.M. Bldg. 3112 W. Highland Blvd, Milwaukee, Wisc. (Submitted by R. W. Schroeder, Sec'y.)

NEW YORK: Sept. meeting—Short film *Lubrication Ain't No Problem,* followed by a Plant Lubrication Panel of four industry representatives discussing the practical side of organizing and operating lubrication departments. Oct. meeting—M. E. Merchant, Cincinnati Mill. Mach. Co. and ASLE President, discussed his experiences with cutting fluids, the relationship with tool angles and materials, and cutting speeds; he further spoke on the major accomplishments and objectives of ASLE. Nov. meeting—R. A. Shartle, Vickers, Inc., spoke on *Trends in Hydraulic System Designs,* outlining past and future thinking in hydraulic system design trends with some references to fluid problems. Future meetings: December 11 (Panel — Recent Lubrication Developments) and Jan. 21 (Diesel Engine Lubrication), 6:00 P.M., Rosoff's Restaurant, 147 W. 43rd St., Manhattan, N. Y. C. (Submited by E. Landau, Sec'y.)

NORTHERN CALIFORNIA: Oct. meeting—Speaker A. G. Zima, The International Nickel Co., Inc., and movie, *Corrosion in Action* in three parts. (Submitted by P. M. Ruedrich, Sec'y-Treas.)

PHILADELPHIA: Sept. meeting—Panel Discussion *Be A Lubrication Expert in Your Own Plant* with panelists J. Dunigan, SKF Industries, L. Miller, Scott Paper Co., and R. Sheetz, Henry Disston & Sons Co., commencing with the basic fundamentals of the job of lubrication engineer, discussion of the proper application of the lubricant, how to reduce the quantity of lubricants in any one plant, and the advantage of a central location with the plant for storage and maintenance of lubricants and lubricating devices. It was decided to form a Clearing House within the section to answer any and all problems on the subject of lubrication of business firms in the area. Nov. meeting—V. E. Amspacher, Pennsylvania Railroad, presented an address entitled *Reclamation of Used Industrial Oils.* (Submitted by J. L. Beatty, Publicity Committee Chairman.)

SAGINAW VALLEY: Future meeting: Jan. 22, 6:45 P.M., Zehnders Hotel, Frankenmuth, Mich., (Submitted by D. H. Buell, Sec'y.)

TWIN CITIES: Sept. meeting—Reorganization meeting with W. P. Youngclaus, Jr., ASLE Administrative Secretary, and J. W. Hopkinson, Penn-Petroleum Corp. and ASLE Midwestern Regional Vice-President, as counselors. (Submitted by W. H. Furst, Sec'y.)

WHEELING: Sept. meeting — Charter Meeting and election of officers (See Directory), with M. E. Merchant, Cincinnati Mill. Mach. Co. and ASLE President, as speaker of the evening discussing *Problems of Friction and Wear*, illustrated with slides, followed by a question-and-answer session. (Submitted by W. P. Liller, Sec'y-Treas.)

YOUNGSTOWN: Future meetings: Dec. 11 and Jan. 8, 6:00 P.M., V.F.W. Bldg., Mural Restaurant, 225 W. Boardman St., Youngstown, Ohio. (Submitted by R. S. Scheidemantel, Sec'y.)

(Patent Abstracts, from 302)

metal soap of fatty acids having between 12 and 24 carbon atoms, a metal salt of an organic acid having a molecular weight below about 160, and a metal soap of a branched chain aliphatic carboxylic acid having between about 10 and about 20 carbon atoms, of which at least 2 carbon atoms are tertiary and having at least one thioether linkage in the chain.

LUBRICATING GREASE COMPOSITIONS, Patent #2,612,473, A. J. Morway and P. V. Smith, Jr., assignors to Standard Oil Development Co. A lubricating grease composition comprising a major proportion of lubricating oil thickened to a grease consistency with a combination of a metal soap of fatty acids having between 12 and 24 carbon atoms, a metal salt of an organic acid having a molecular weight below about 160, and a metal soap of a branched chain aiphatic carboxylic acid having between about 19 and about 20 carbon atoms, of which at least 2 carbon atoms are tertiary and having at least one ether linkage in the chain.

KETO-ACID SOAP GREASES, Patent #2,613,182, L. W. Sproule and L. F. King, assignors to Standard Oil Development Co. A lubricating grease composition comprising a major proportion of lubricating oil thickened to a grease consistency with a minor proportion of a metal soap of 12-keto stearic acid.

SULFURIZED ADDITIVES FOR LUBRICATING COMPOSITIONS, Patent #2,613,183, A. D. Kirshenbaum and J. M. Boyle, assignors to Standard Oil Development Co. An extreme pressure additive composition for lubricants consisting essentially of the reaction product of about 1 molar proportion of a monocyclic terpene with about 1 to 2 molar proportions of a sulful chloride, the quantity of said sulfur chloride being substantially sufficient to saturate one double bond in said terpene, said reaction product being further reacted with about 0.001 to 0.1 molar proportion of phosphorus sulfide at a temperature of 100 to 175 C.

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★ Filtering hydraulic oil eliminates sticking and clogging of valves and reduces down time and maintenance cost. A prominent authority on hydraulic machinery says that 70% of servicing and repairs is due to the improper condition of hydraulic oil.

★ Filtering coolant fluid removes all fine metal particles and grit from grinding wheels that is pumped back to nozzles unless a filter is employed in cleaning up the return coolant.

★ Clean filtered coolant prevents scratching and flat spots on work piece and increases the life of grinding wheels and cutting tools. Reduces the frequency of dressing grinding wheels.

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LUBRICATING OIL CONTAINING A COPOLYMER OF AN ESTER OF AN UNSATURATED ACID AND A N-HYDROCARBON AMIDE OF AN UNSATURATED ACID, Patent #2,613,184, W. E. Catlin, assignor to E. I. duPont deNemours & Co. A lubricating oil containing from 0.1% to 10% by weight of an oil-soluble copolymer of an alkyl ester, in which the alkyl ester group contains from 4 to 18 carbon atoms, of an unsaturated aliphatic acid, and a N-hydrocarbon amide of an unsaturated aliphatic acid in which the hydrocarbon radical attached to the nitrogen atoms has from 4 to 18 carbon atoms, said unsaturated acid in each instance being selected from the group consisting of acrylic acid and alpha-alkylacrylic acids in which the alkyl group contains from 1 to 3 carbon atoms, the amide-nitrogen content of said copolymer ranging from 0.2% to 7.0% by weight.

GREASE COMPOSITIONS, Patent #2,614,076, R. J. Moore and W. Saarni, assignors to Shell Development Co. A grease composition comprising a lubricating oil base and a lithium soap of a hydroxy fatty acid having 10 to 24 carbon atoms, wherein the hydroxyl group is more than 9 carbon atoms removed from the carboxyl group, in an amount sufficient to thicken said oil to a grease consistency, and a second alkali metal soap of an organic monocarboxylic acid having 3 to 18 carbon atoms, said second soap bearing a substituent which is capable of forming a co-ordinate bond and which is less than 9 carbon atoms removed from said carboxyl group, the second soap being present in an amount between 0.1% and 0.25% by weight of the grease composition, the substituent being one selected from the group consisting of hydroxy, keto, amino, nitro, mercapto, thiol and sulfo groups.

GREASE COMPOSITION, Patent #2,614,077, R. J. Moore and W. Saarni, assignors to Shell Development Co. A grease composition containing only carboxylic acid soaps as gelling agents comprising a lubricating oil, 4 to 25% by weight of a mixture of lithium soap of an aliphatic monocarboxylic acid having 10 to 24 carbon atoms bearing no polar groups other than the carboxyl group, and an alkali metal soap of an aliphatic monocarboxylic acid having 3 to 18 carbon atoms bearing a hydrogen bonding substituent less than 9 carbon atoms removed from the carboxyl group, the second soap being present in an amount between 4 and 12 mole per cent of the total soap content, said hydrogen bonding substituent being one selected from the group consisting of hydroxy, keto, amino, nitro, mercapto, thiol and sulfo groups.

LUBRICATING COMPOSITIONS, Patent #2,614,078, L. R. Churchill, assignor to Tide Water Associated Oil Co. In the preparation of lubricating compositions by blending a mineral oil with a sulfurized fatty oil that substantially increases the color characteristics of said mineral oil, the improvement which comprises subjecting said blend to controlled contact with a free-oxygen containing gas in the presence of a metal salt of an organic acid possessing activity as an oxidation catalyst for a period of time sufficient to substantially decrease the color characteristics imparted to said blend sulfurized fatty oil.

LUBRICATING GREASE, Patent #2,614,079, R. J. Moore, assignor to Shell Development Co. A grease composition comprising a major amount of a mineral lubricating oil having incorporated therein in an amount sufficient to form a grease, an anisometric fibrous lithium soap, said fibers being a mixture of fine and coarse fibers, respectively, in the ratio of from 3:1 to 1:3.

LUBRICATING OIL ADDITIVE MATERIALS, Patent #2,614,080, L. M. Welch, assignor to Standard Oil Development Co. A lubricating composition consisting essentially of a major proportion of a waxy lubricating oil base having combined therein 0.01% to 10.0% by

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weight, based on the weight of the total composition, of an alkylated polystyrene which has been prepared by first reacting polystyrene having a molecular weight within a range of from 10,000 to 70,000 with an approximately equimolar proportion of a linear aliphatic monoolefin having from 12 to 22 carbon atoms in the presence of a Friedel-Crafts catlyst and reacting the product thus formed with approximately 2 molar proportions based on the polysterene of a linear aliphatic mono-olefin having from 2 to 10 carbon atoms in the presence of a Friedel-Crafts catalyst to obtain a final product having a molecular weight within a range of from 7,000 to 20,000.

LUBRICATING COMPOSITION CONTAINING BORIC ACID, Patent #2,614,985, D. L. Cook, assignor to Shell Development Co. A lubricating composition comprising at least 90% of a mineral lubricating oil containing dispersed therein from about 0.05 to about 2 wt. per cent of boric acid.

GREASES BASED ON FINELY DIVIDED INORGANIC MATERIAL, Patent #2,614,986, A. Beerbower and A. J. Morway, assignors to Standard Oil Development Co. A solid grease composition consisting essentially of a mineral base lubricating oil of 35 to 1000 SSU viscosity at 100° F. thickened to a grease consistency with 10 to 25% by weight of a finely divided inorganic material having a bulk density of less than 2 lbs./gallon in its normal finely divided dry condition, said inorganic material being selected from the group consisting of magnesium oxide and magnesium carbonate.

GREASE COMPOSITIONS, Patent #2,614,987, J. L. Dreher and C. F. Carter, assignors to California Research Corp. A grease composition consisting essentially of a major portion of a lubricating oil, a thickening agent, and from about 0.1% to about 5%, by weight, of a polyvalent metal dithiocarbamate, and from about 0.01% to about 2%, by weight, of a dihydroxyanthraquinone.

HYDROCARBON OILS CONTAINING S-ALKOXYMETHYL — O, O' — DIALKYLDITHIOPHOSPHATES, Patent #2,614,988, E. O. Hook and P. H. Moss, assignors to American Cyanamid Co. A hydrocarbon oil composition comprising a major amount of a hydrocarbon oil having dissolved therein, in a sufficient amount to prevent deterioration by oxidation, a condensation product.

LUBRICATING COMPOSITION OF MATTER, Patent #2,614,990, D. Harman and R. E. Thorpe, assignors to Shell Development Co. A composition of matter comprising a major amount of a neutral liquid vehicle and a minor amount sufficient to impart extreme pressure properties to said vehicle of a mixture of phosphorus compounds.

STEAM ENGINE LUBRICANTS, Patent #2,614,991, J. I. Wasson, assignor to Standard Oil Development Co. An emulsible, soap-free steam cylinder oil composition consisting essentially of about 87% by weight of predominantly Mid-Continent mineral lubricating oil of about 135 to 300 SSU viscosity at 210 F., about 10% of a resinous propane precipitated Pennsylvania type paraffinic petroleum oil residue of substantially higher viscosity, and about 3% of tallow.

Prepared by ANN BURCHICK from Official Gazette—Vol. 661, No. 4; Vol. 662, Nos. 1, 2, 3, 4, 5; Vol. 663, Nos. 1, 2, 3. Printed copies of patents are available from the Patent Office at twenty-five cents each. Address the Commissioner of Patents, Washington, D.C., for copies and for general information concerning patents.

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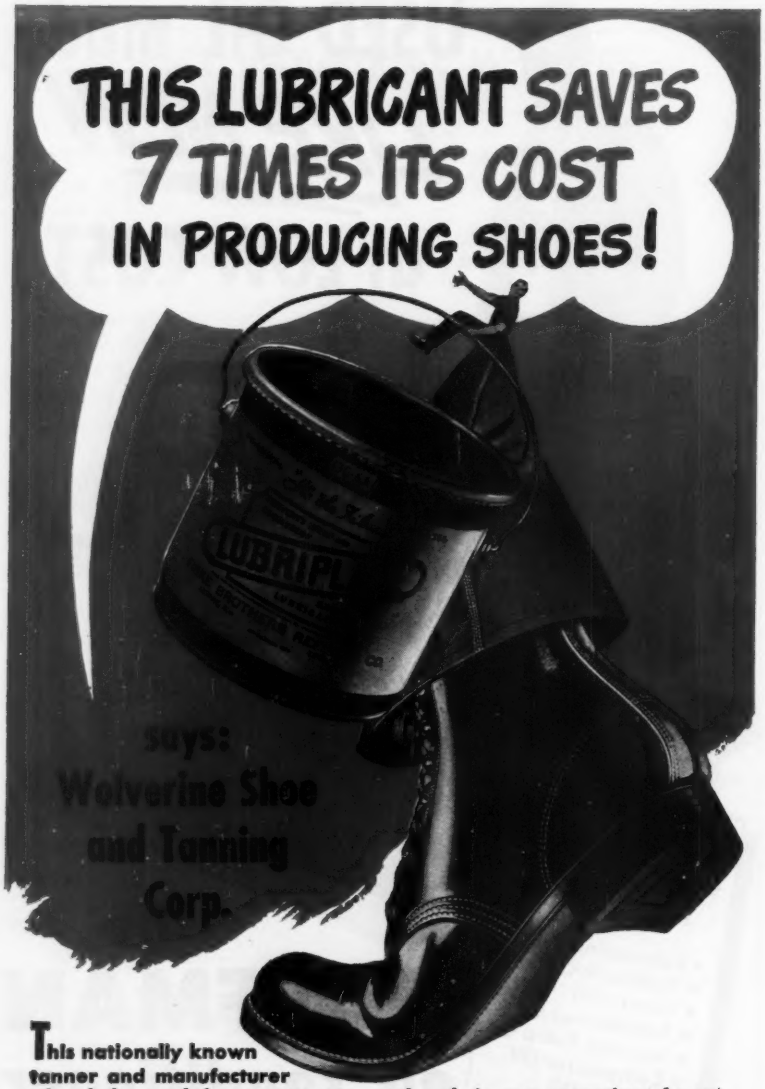

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RESEARCH AND PROJECTS: The American Society of Lubrication Engineers sponsors, with the support of industry, research projects which will be of benefit to the lubrication engineering profession. The Research and Projects Committee evaluates the need for and the scope of such projects and makes the necessary working arrangements. At present a literature survey on bearing design and its effect on lubrication is undertaken by Columbia University, New York, as an ASLE sponsored research project.

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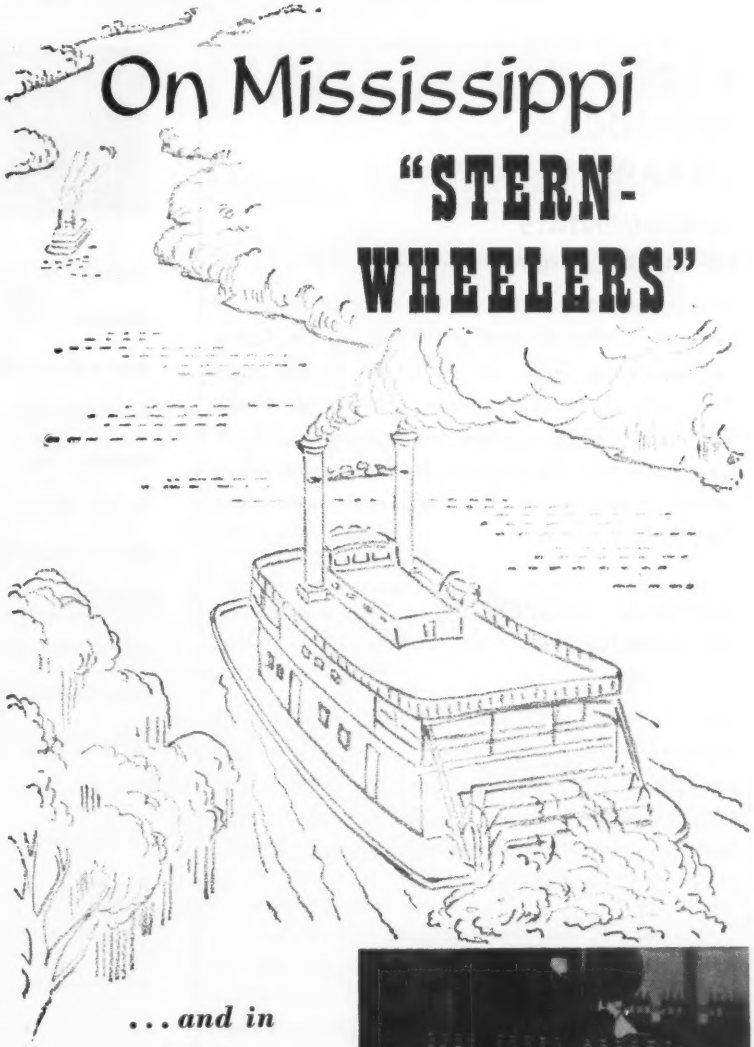

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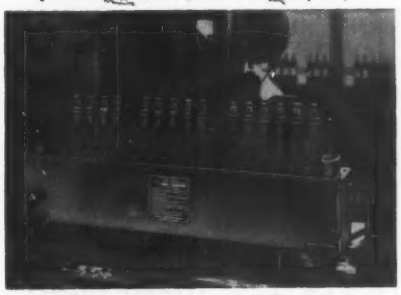

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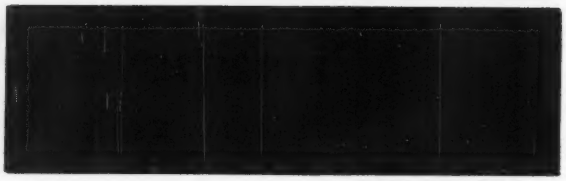

Acheson Colloids Co. 274
Alemite 279
Alpha Corp., The 314
Archer-Daniels-Midland Co. 321
Bowser, Inc. 317
Brooks Oil Co., The 312
Burns Laboratories, Inc. 305
Classified 308
Climax Molybdenum Co. 320
Dow Corning Corp. 315
Farval Corp., The . . . . . (Inside Back Cover)
Gits Bros. Mfg. Co. 316
Grafo Colloids Corp. 306
Hodson Corp., The . . . . . (Back Cover)
Honan-Crane Corp. 275
Ironsides Co. 278
Lincoln Engineering Co. 325
Lubriplate Div., Fiske Bros. Refining Co. 324
Madison-Kipp Corp. 311
Manzel, Inc. 327
Morgan Construction Co. 276
Norgren, C. A. Co. 318
Shell Oil Co. 326
Sinclair Refining Co. 310
Sparkler Mfg. Co. 319
Stuart, D. A. Oil Co., Ltd. 273
Sun Oil Co. 307
Tide Water Assoc. Oil Co. 277
Trabon Engineering Corp. . . . . (Inside Front Cover)
Trico Fuse Mfg. Co. 313
U. S. Hoffman Machinery Corp. 323

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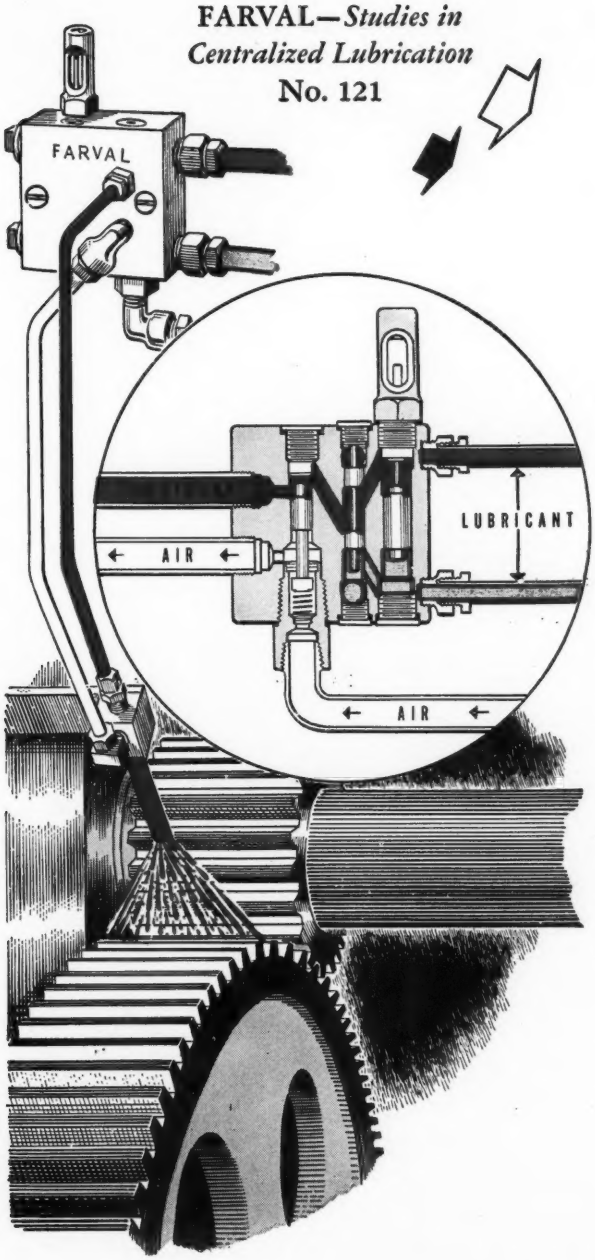

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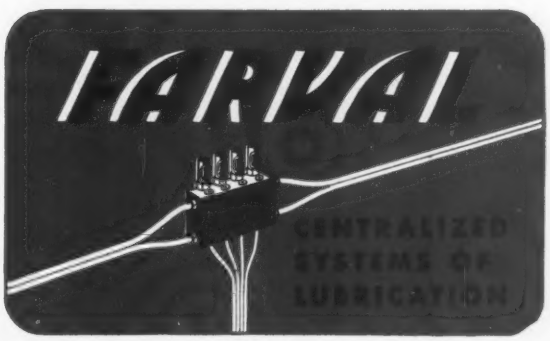

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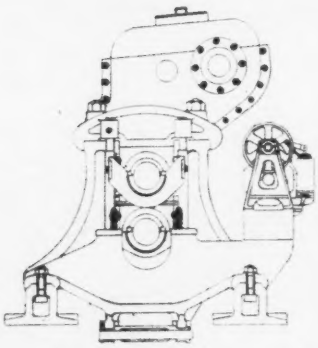

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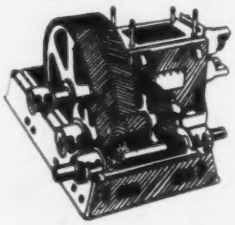
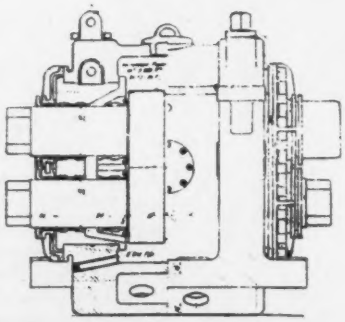
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